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THE DESIGN AND CONSTRUCTION OF BALL BEARINGS.

C. H. BENJAMIN.

The use of steel balls for reducing the friction of bearings is being rapidly extended, and new applications are daily being made. The fact that one company alone have a monthly output of from fifteen to twenty millions of balls, ranging in size from 1-16 inch to 4 inches, is a sufficient index of the importance of this relatively new branch of machine design.

The development of the bicycle has been one reason for the phenomenal demand for steel balls, but what is sauce for the goose is sauce for the gander, and what is good for bicycle bearings has been found good for bearings in other machines.

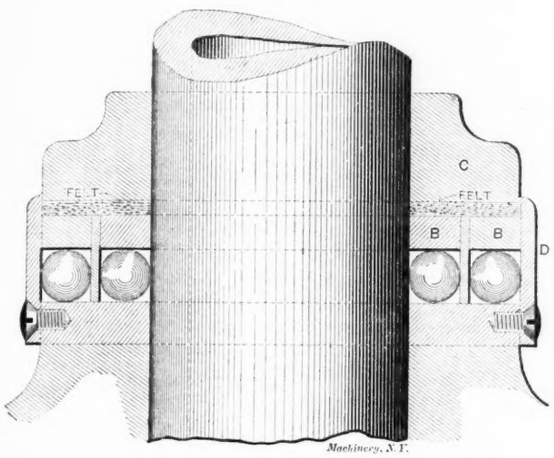


FIG 5.

One thing still stands in the way of the development of this branch of design, and that is the almost entire absence of accurate knowledge in regard to the strength and durability of steel balls, and the proportions that should obtain under given conditions of pressure and speed. When the editor of MACHINERY suggested to the writer the preparation of an article on this subject, it seemed to be a case of bricks without straw.

Now, the first name that comes into one's mind when the subject of ball bearings is mentioned is that of John J. Grant. Acting on this thought, and hoping for straws at least, the writer called on Mr. Grant at his office in the works of the Cleveland Machine Screw Co., and was hospitably received by that gentleman.

A statement of the object of the call elicited from Mr. Grant the rather unexpected and decidedly depressing information that he himself did not know how to proportion ball bearings, and further, that he believed no one else did.

After recovering somewhat from the shock of this communication, the writer despairingly asked Mr. Grant how he determined the size and number of balls for any given set of conditions.

In reply Mr. Grant told a story of his boy, who, at a comparatively early age, undertook to make a banjo. The young constructor succeeded fairly well until he came to string the instrument. He had no preconceived ideas on the subject of tuning, and so he tightened up the first set of strings until they broke, when he had reliable experimental data, and consequently screwed the next set up not quite so far.

"So," said Mr. Grant, "we get our data by experience, and make the next bearing a little larger."

He further remarked that the company with which he is connected has had a large number of tests made on the strength of steel balls, but that the tests were not altogether satisfactory, as they disclosed a considerable variation in the compressive strength of balls made by the same process.

There is great need of further experiments on the strength and endurance of steel balls under conditions approximating those obtaining in practice.

The writer desires to express his thanks to Mr. Grant and to Mr. Clyne, the superintendent of the Cleveland Machine Screw Co., for their courtesy in putting at his disposal all the material in their possession which would aid in the preparation of this article, and to say that the most of the bearings illustrated are from designs furnished by them.

STRENGTH.

In 1895 quite an extensive series of tests on the compressive strength of steel balls was made at the laboratories of Sibley College for the company above mentioned.

An abstract of the results may be found in Riehle's Digest of Physical Tests for January, 1896.

The principal tables given are here reproduced for reference. Table I. shows the strengths of different sizes for balls under a crushing load, the balls being placed between hardened steel plates:

Table II. shows the variation in the crushing strength of ¼-inch balls, due to differences in the steel and in the process of

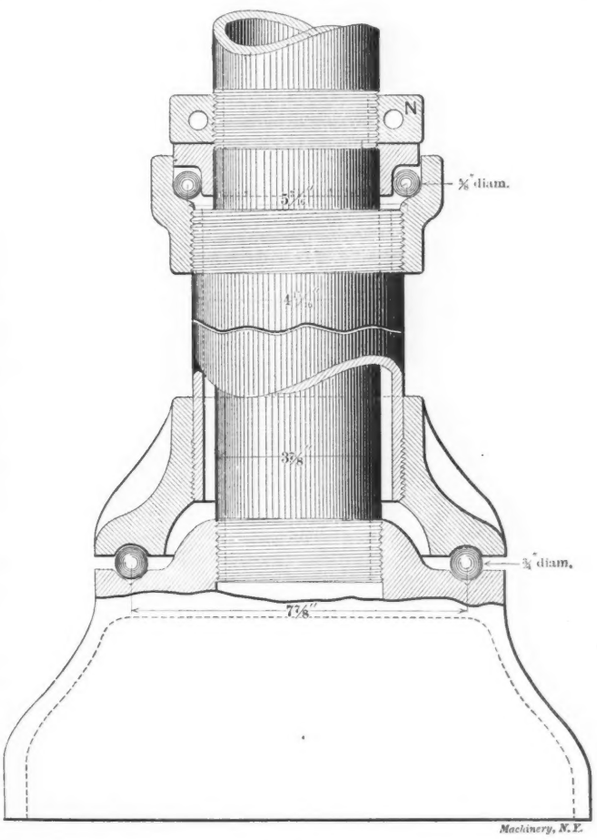


FIG. 6.

hardening, and also shows the depth of indention of the hardened steel plates.

The results in Table I. show a lack of uniformity in the increase of strength with size. This may be due to the fact that only a very limited number of some of the sizes were tested, thus not giving an average.

Some similar tests have been made during the present year

at the mechanical laboratory of the Ohio State University, and a synopsis of the average results are given in Table III.

TABLE NO. 1.

No.	Diameter.	Maximum Crushing Load. Average of Three Tests.	Minimum Crushing Load. Average of Three Tests.	Number of Sets of Three.
1	¼ inch.	10 683	5 566	21
2	⅛ "	3 500	2 923	3
3	½ "	11 026	7 460	3
4	⅜ "	17 690	.....	1
5	½ "	39 766	.....	1
6	¾ "	30 426	23 990	3
7	1 "	55 210	.....	1
8	2 "	220 000	Only 1 ball tested.	.....

TABLE NO. 2.

No.	Lot 1, Kidd's.		Lot 2, Kidd's.		Cresent.		Jonas & Colvin.	
	Pounds.	Indent.	Pounds.	Indent.	Pounds.	Indent.	Pounds.	Indent.
1	7 495	.0084	7 512	.0009	8 195	.....	6 410	.0055
2	9 905	.0090	6 905	.005	6 960	.0057	6 725	.0096
3	4 485	.0036	7 300	.001	6 000	.0065	7 080	.0019
3	8 010	.0044	8 520	.0014	9 000	.0099	6 250	.0057
5	6 000	.0039	6 790	.0017	8 995	.0078	6 305	.0031
6	6 880	.0018	8 370	.001	9 460	.0015	6 510	.0065
7	6 715	.0072	7 365	.0044	8 515	.0062	6 510	.0061
8	7 500	.0037	6 977	.0014	6 675	.0055	4 000	.0038
9	6 265	.0010	7 283	.001	7 770	.0058	6 965	.0030
10	6 840	.0059	6 735	.0061	8 740	.0060	6 510	.0081
Ave.	7009 5		7375·7		8 031		6326.5	

TABLE NO. 3.

Diameter.	Average Crushing Load.	No. of Balls Tested.
⅛	1 772	5
¼	6 338	6
⅜	14 799	13
½	33 132	17
⅝	40 847	20
¾	61 616	3
⅞	60 800	3
1	77 000	3

The ⅞ and 1-inch balls in the above table were made of a different steel, which probably accounts for the relative diminution in strength.

Roughly speaking, in designing a bearing where no other data

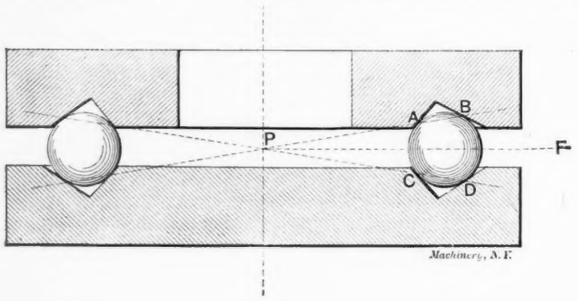


FIG. 1.

are at hand, a factor of safety of five is generally used, taking the average strength from the tables. For instance, ¼-inch balls may be allowed a working pressure of 1,200 to 1,500 pounds per square inch, if the speed is not excessive.

FAILURE OF BALLS.

Steel balls in bearings do not fail by crushing, and for this reason the rule just given can only be considered as an approximate one.

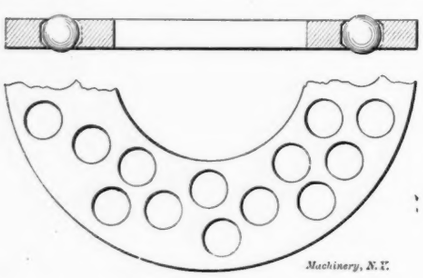


FIG. 3.

The continual rolling of a ball under pressure gradually stretches or draws the exterior surface, making the coat too large for the wearer, until a scale or shell separates from the center and the usefulness of the ball is gone. The life of the ball under this treatment is a function of both speed and pressure; if either is excessive the life is short.

The remedy is to increase the number of balls and thus subdivide the pressure, or to run the bearing at a lower speed. Ball bearings have been much more satisfactory at comparatively slow speeds.*

If a bearing is not kept clean, the presence of dust or grit may cause the balls to wear out, but the usual method of failure is that indicated above. It may be said of a ball bearing as of toothed gearing, that the failure of a unit usually precipitates a catastrophe.

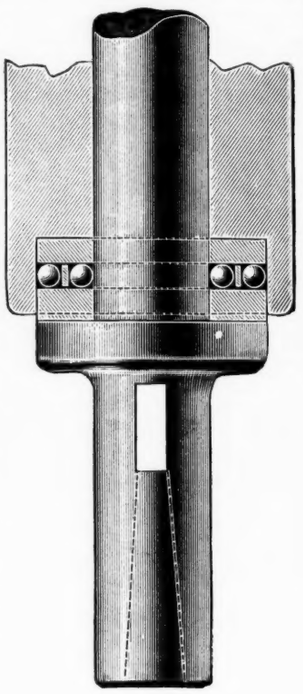


FIG 2.

In general, with a given diameter of ring, an increase in the size of balls, with a corresponding decrease in the number, will give a stronger and more durable bearing.

Of course there is practically a limit to this, but it is difficult to say where the limit may be.

For example, we will suppose the ball circle to be $2\frac{9}{16}$ inches in diameter, or about 8 inches in circumference. Thirty-two ¼-inch balls would have a safe crushing strength of about $\frac{7,500}{5} \times 32 = 48,000$ pounds. Sixteen ½-inch balls, on the other hand, would fill the same circle, and have a safe crushing strength of about $\frac{30,000}{5} \times 16 = 99,000$ pounds, the strength being approximately proportional to the projected area, and probably the durability about the same.

DESIGN OF BEARINGS.

One of the first questions that confronts a designer is that of patented devices. He asks, "What devices can I use that will not conflict with existing patents, and that yet will be practicable and desirable?" The answer to this query is that the field

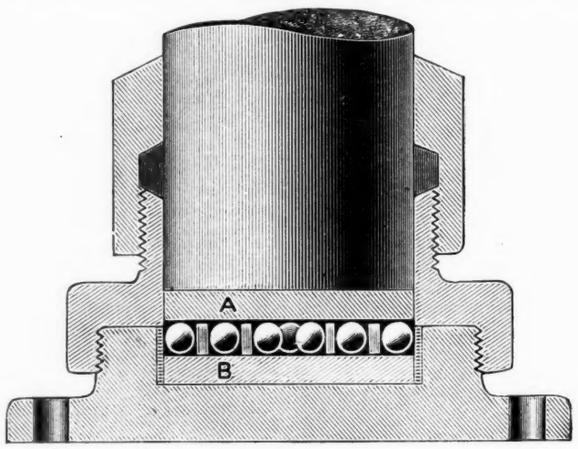


FIG. 4.

is practically open to all comers. The use of the balls themselves is old and has long been public property.

The separating rings and cages, the grooves and collars can be used without fear of infringement.

It is safe to say that many of the existing patents on such devices are worthless and could easily be broken if occasion required. Several experts with whom the writer has conversed

* One bearing of special accuracy has been made to run at 30,000 revolutions per minute, without exeessive vibration, but the pressure in this case was almost nil.

are of this opinion. The numerous designs shown in this article are all in use, and are free to the public.

In order to insure perfect rolling contact the bearing points or surfaces should be so designed that there be no slipping at any point, and the balls must be of exactly the same diameter.

The collars, disks or cones used as bearings for the balls, should be of hardened steel, preferably ground after hardening to perfect truth.

END BEARING OF THRUST JOURNALS.

Ordinarily end bearing journals or collars have been a source of unlimited trouble, since the velocity of rubbing varies as the distance from the center and the wear is correspondingly uneven. Centrifugal force causes the lubricant to leave the center of the pivot and cutting frequently ensues.

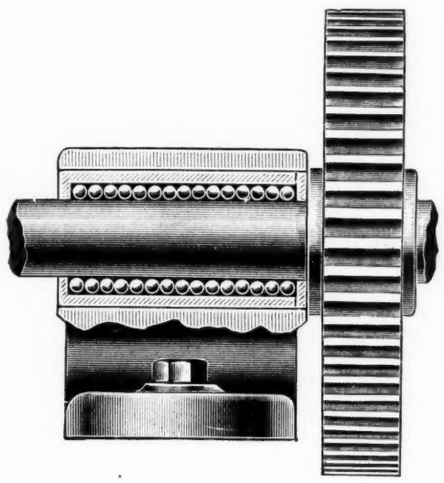


FIG 7.

The use of one or more zones of steel balls does away with these difficulties, and at the same time substitutes rolling for sliding friction. A bearing of this kind may have one or more circles of balls running between flat plates with separating rings, as in Fig. 2, or the balls may be run in grooves, as shown in Fig. 1.

In the latter case, if there are four points of contact, the groove must be laid out as shown in the figure. The touching points on opposite sides of the ball, as *A* and *B* or *C* and *D*, must lie in the straight lines radiating from the center of the axle at *P*.

The ball will then continually turn about the axis *P F*, and the points *B* and *D* on the ball will move faster than *A* and *C* in the same ratio as the corresponding points on the revolving collar, so that there will be no slipping at either point.

The action will be similar to that of a cone *P B D* rolling between two other cones, as in box gearing.

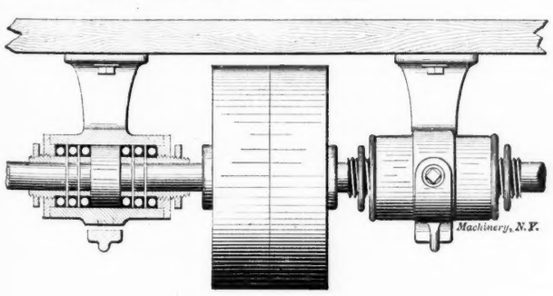


FIG. 8.

Fig. 2 shows the application of balls and flat collars to the spindle of a drill press. The relative proportions of spindle, balls and collars can readily be estimated from the figure.

The use of cages to hold balls in place is usually unnecessary, and introduces an additional element for friction. In case, however, it is necessary to remove the bearing at frequent intervals, some such device will prevent scattering and loss of the balls. A very ingenious cage was recently shown the writer by Mr. Clyne, and is illustrated in Fig. 3.

A ring of bronze, of a thickness slightly less than the diameter of the balls, has a series of holes drilled irregularly in its flat side, so that the balls will be an easy fit. The holes are drilled not quite through, but just enough burr is left to prevent the balls dropping through. The balls are then placed in position and the metal around the open ends of the holes slightly upset by tools adapted to the purpose. This last operation is quickly performed in an ordinary press, and prevents the balls from escaping, while leaving them entirely free to turn.

It is better if the balls are so arranged in the surface of the rings as to bear at all distances from the center and prevent the formation of ruts or grooves in the bearing collars.

This ring can be slipped on a shaft or spindle between the two hardened steel collars, the same as the ordinary loose friction

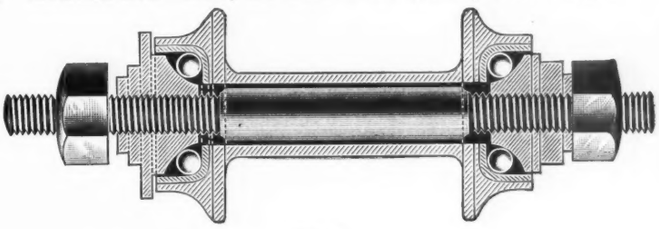


FIG. 9.

ring. Even if the holes wear they will never wear in such a way as to release the balls.

In order that a ball bearing may have a long life and do good service, it is absolutely necessary that it be kept clean and free from dust.

It may or may not be oiled; doctors disagree on that point, but cleanliness is indispensable. Probably a slight greasing of the balls with vaseline when first put in will do away with any necessity for oiling. Fig. 4 gives a vertical section of a step bearing, designed for a pulverizing mill, where the quality and quantity of dust was such as to speedily destroy the bearings, unless properly protected. The spindle in this machine is 7 inches in diameter, and rests on three rows of 7/8-inch steel balls, separated by rings of machinery steel 13-16 inch wide and 3/8 inch thick.

The bearing plates *A* and *B* are made of tool steel, hardened and ground, and are each 1 inch thick. The bearing is protected from dust by the stuffing-box and gland enclosing a

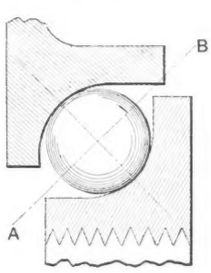


FIG. 10.

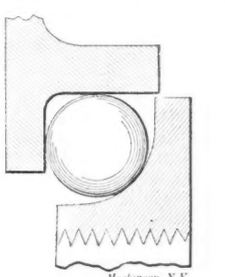


FIG. 11.

packing of felt. The proportions of the stuffing-box and of the step can be estimated from the figure.

Fig 5, page 69, shows a dustproof bearing where a washer of felt is interposed between the collar *C* and the bearing rings *B B*. The stationary guard ring *D* comes in close contact with the felt, and effectually excludes dirt. In this design the spindle is 4¾ inches in diameter, and is supported by two rows of 15-16 balls, there being 27 in the outer and 19 in the inner row. The bearing rings are 5/8 inch thick above and ¾ inch thick below the balls.

In Fig. 6 is shown a design, which would be appropriate for a revolving sleeve exposed to both vertical and side thrusts, the lower ring of balls taking care of the vertical and the upper ring of the lateral pressure. The nut *N* allows of adjustment for the upper bearing. Dimensions are given in the cut.

CYLINDRICAL BEARINGS.

The use of balls is not confined to the type of bearings just described, although possibly more of a necessity there than in bearings of the cylindrical shape. When the ordinary shaft or axle is subjected to great pressure, the difficulty of insuring good lubrication has frequently led to the adoption of steel balls,

as shown in Fig. 7. If one asks as to the number of balls to be used in such a case, the answer would be, as many as the box will hold.

For a lighter bearing, running at a higher speed, it would be better to use fewer balls and employ separating rings, as shown in Fig. 8.

If a journal is exposed to a certain amount of end thrust as well as lateral pressure, it is customary to employ cone bearings, the elements of the cone being approximately at right angles to the direction of the resultant pressure.

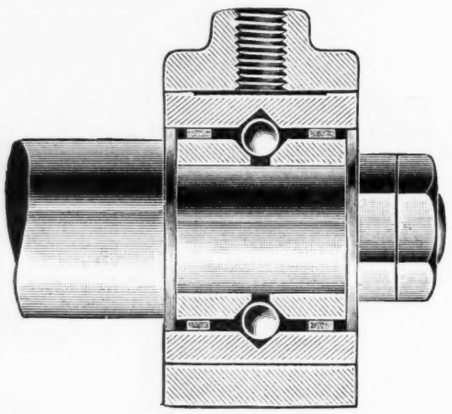


FIG. 12.

The most familiar example of this is probably the bicycle hub bearing, as shown in Fig. 9, and which needs no explanation. There has been considerable discussion as to the proper shape of the bearing surfaces and the number of points of contact.

The best practice is to confine the bearing to two points on each ball, to have these points at opposite extremities of a diameter and to have the curvature of the cones slightly less than that of the ball. See Fig. 10.

The ball is then held in a definite position, has a definite axis, *A B*, and rolls without slipping.

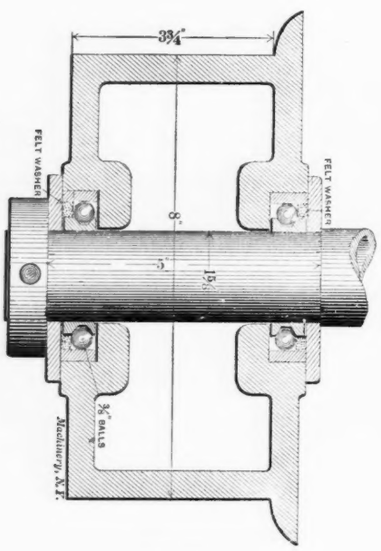


FIG. 13.

If there are three points of contact, as shown in Fig. 11, there must be slipping somewhere, and the rolling will be irregular. This fact has been demonstrated repeatedly in bicycle bearings.

MISCELLANEOUS APPLICATIONS.

Fig. 12 shows a bearing which is well adapted for light pressures at moderately high speeds. The cone-shaped rings will preserve the relative positions of shaft and box, while the felt strips *F F* exclude the dust and retain the lubricant, if any is used. A bearing similar to this has been applied to wheel barrows for hauling brick, and has noticeably increased the amount of work which the laborer can do in a day.

The loose pulley or truck has always been a source of vexation, on account of the difficulty of making the oil move against centrifugal force. This is the ball bearings' opportunity, and Fig. 13 shows how well it has been improved. The balls are ⅜ inch in diameter, and are held in place by the groove on the inner ring, while a felt washer again keeps out the dust. The rings are ⅝ inch wide, and 3-16 inch thick. The principal dimensions are shown in the figure.

In Fig. 14 is illustrated a double thrust bearing designed for the screw which drives the carriage of a plate planer. There are two rings of ⅜ inch balls in each bearing, 46 in the outer ring and 36 in the inner. The weight of the screw in this case is carried by the babitted bearing in the center, the balls being intended to take the end thrust only.

Fig. 15 shows what may perhaps be considered the most ingenious application of balls yet made—a ball-bearing screw and nut. The balls follow each other, like sheep through a hole in the wall, traversing all the windings of the screw and finally returning through the bye pass only to begin over again their interminable wanderings. This has been used in the Sprague elevator.

The thrust of the screw shaft in heavy worm gearing, such as is used in hoisting machinery, should be taken by ball thrust bearings, as this will increase both efficiency and durability.

To describe the various applications already made would consume too much time and space. Steel balls are often used where we would least expect them. The manufacturers of agricultural machinery and implements have found in them a bearing which excludes dirt and reduces friction without oiling.

A short list of some of the applications of ball bearings, besides those already mentioned, would include propeller shafts, cranes, turn tables, step bearings for water wheels, jack screws, gun carriages, lighthouse lanterns, observatory domes, trolley wheels, well drilling machinery, etc., etc.

SUMMARY.

In closing it may be well to sum up our somewhat meager information as to the principles governing the design and construction of this species of bearing.

1. The ball must be capable of rolling indefinitely without bringing up—*i. e.*, the chain of the balls must be endless. Notice

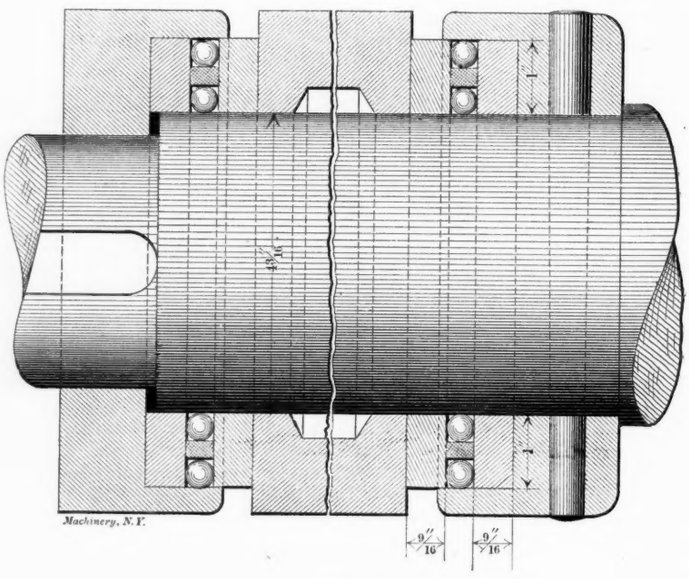


FIG. 14.

the application of this in Fig. 15; if the path for the return of the balls had not been provided, the bearing would be absolutely useless. It may seem superfluous to call attention to this fact, but several devices have actually been patented in which this principle was entirely disregarded.

2. The balls must roll in a definite path, and about some definite axis, and not slip at any of the points of contact. This has been explained in Figs. 1, 10 and 11. Some designers seem to think that the balls may be flung in at random and allowed to play a go-as-you-please race.

3. The balls must be spherical and of exactly the same size.

If one ball in the bearing departs from these conditions, it affects the action of all the others.

4. Be cautious about using balls in high speed bearings, as there is a limit to their usefulness in this direction. They show to the best advantage where the speed is slow.

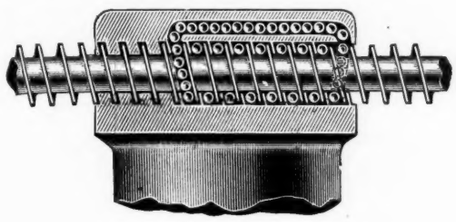


FIG. 15.

5. Use judgment in regard to the number and size of balls under given conditions, fortified by your own experience and that of others, since no rigid rules can be given.

6. Do not try to use ball bearings on everything, since they are no more a universal panacea than electrictiy or triple expansion.

It is usually better in machine design not to be too radical. Let the other fellow do that.

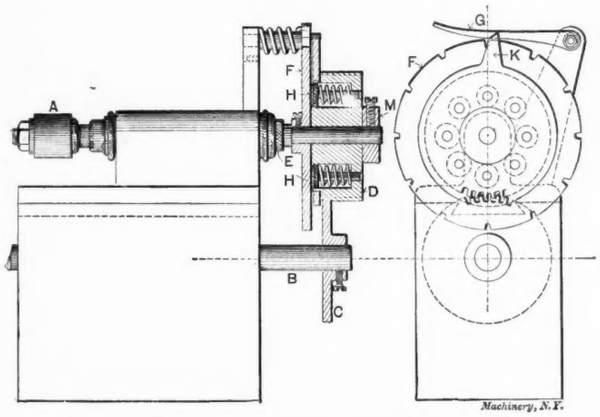
* * *

AN AUTOMATIC INDEXING DEVICE.

G. P. WADLEIGH.

In the wheel cutting device illustrated, A is a number of wheels or gears to be cut, B the main shaft carrying the cam to give the feed motion and the gear C in mesh with the gear D, which is loose on the quill E.

F is the index plate fast on the quill. The gear D has a number of rawhide plungers, H H H, which are pressed against the index plate by the springs behind them, so that as the gear D is constantly turning, the friction of the rawhide roller tends to turn the index plate and quill. The index plate, however, is prevented from turning by the pawl G until the projection K on the gear D releases the pawl, when the quill is allowed to turn until the pawl drops into the next notch in the index.



There are two advantages in this device; first, that it could be applied to cases where the indexing is irregular, as in drilling watch balances, where I think it was first used; second, that it requires no cams or complication of levers, as is usual, and no adjustment for different numbers of teeth. All that is necessary to change from cutting one number of teeth to another is to remove the collar M, gear D, and put on a new index plate.

* * *

THE SHOP COMMITTEE.

The system of the National Cash Register Co., of Dayton, O., seems to be attracting considerable attention lately, and everybody is talking about the "Shop Committee."

The Cash Register Co. is not the first concern to have a shop committee, as I know of one that was organized over twenty years ago. The Cash Register Co., however, need not feel at all cut up at learning this, as their "Shop Committee" seems from all accounts to be a success, whereas the other shop committee which I have in mind did not reach that degree of success sometimes described as "howling," as will easily be discovered from the following:

Aaron Artichoke was the superintendent of the factory, and John Juicedrinker was Aaron's "head push," and one day Aaron said: "John, I think it would be a good thing to have the head men of this factory get together once a week for consultation; each one could say what he had seen going wrong in the factory during the week, and criticise the other's faults and peculiarities, everything being done in a friendly spirit. What do you think of it?"

"JOHN, YOU LET THINGS HANG BY THE GILLS."

"Well, Mr. Artichoke," said John, "I think it would be a good plan; none of us are perfect, and I, for one, am willing to stand any proper criticism, and do all I can to improve myself."

"Good," said Aaron; "that's the kind of talk I like, and as long as you like the idea, you and I'll have a little informal meeting right now.

"Your chief trouble, John, is letting things hang by the gills; you get a thing along about so far, call it all right and then leave it to take care of itself, in consequence of which it don't ever get done. Now, John, isn't that so?"

"Yes, Mr. Artichoke," said John, "I know that is my failing; I continually try to overcome it, and will now make extra effort and see if I cannot correct it altogether."

"That's right, John," said Aaron, "I knew you would be reasonable about the matter. Now, what do you consider to be my worst failing?"

NOT BY A — ! — !! — !!!

"Well, Mr. Artichoke," said John, "I think that your failing is about the same as mine—letting things hang by the gills."

"Not by a horse's mother's sight!" said Aaron.

* * * * * *

Note: The first regular meeting of the shop committee was never held.

* * * * * *

P. S.—I hope that the Cash Register Co.'s committee never "scrap." BEEN THERE.

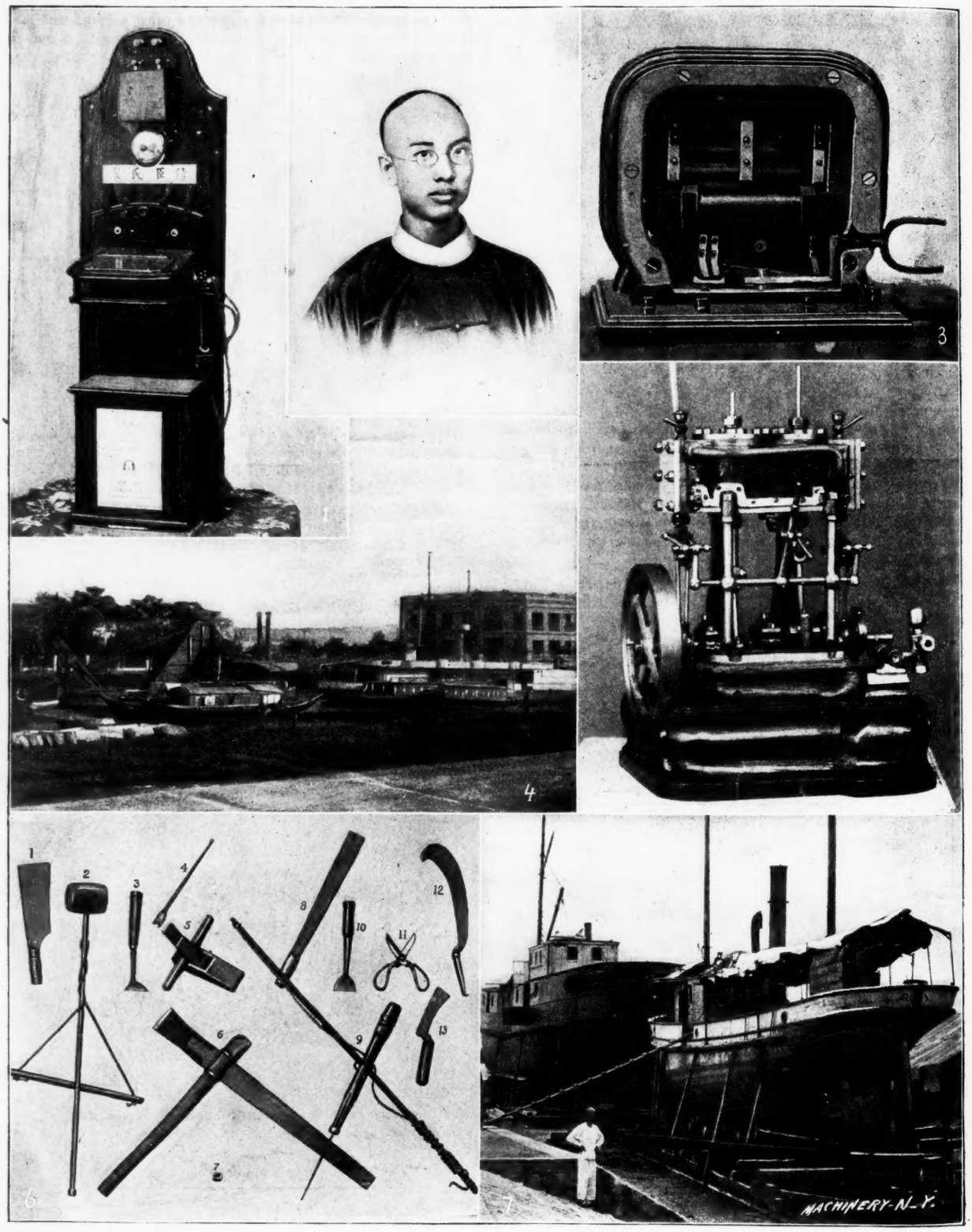
* * *

We call attention to an error in solution No. 1 of Mr. Rogers' chuck problem in our October issue. The figures 40° 26′ should read 44° 26′.

ALL THE WAY FROM CHINA.

We are indebted to one of our subscribers, Mr. Im Shun San, of Haiphong, Tonkin, China, for the interesting group of photographs on this page. "Im Shun San, Esq." is by his own description, "electrician, photographer, machinist and employee at Marty & d'Abbadie's ship building yards," and is certainly a shining example of Chinese progressiveness. That he has a clever and inventive mind is shown by the fact that he himself made the telephone, views of which he has sent us, and built the small marine engine. His mastery of our language is quite remarkable, and his penmanship would do credit to one of our countrymen.

We give the titles of the photographs in Mr. Shun San's own words:



No. 1 represents a Telephone made by Im Shun San. No. 2. Im Shun San, Esq. No. 3. Inside of Shun San's home-made telephone. No. 4. View of Porchet & Co.'s works, Haiphong, Tonkin. No. 5. ½-HP. compound marine engine, made by Shun San. No. 6 shows a collection of Chinese and Tonkin

wood and metal workers' tools, and Mr. Shun San has described the different articles as follows: Nos. 1, 8, 12—Carpenters' chopping and splitting knives. Nos. 3 and 10—Carpenters' chisels. No. 6—Carpenter's adze. Nos. 5 and 13—Wood planer and stiff saw. No. 9—Brace bit. No. 4 and 7—Bit and ring for No. 9. No. 2—Machinist's drill. No. 11—Tinsmith's shears; brass handles and iron blades. This very unique set of tools would probably fill the average American workman with dismay should he suddenly find himself deprived of all others, but our Chinese friends seem to manage very well with them.

The title written under the last picture, No. 7, runs thus: "Marty & d'Abbadie's patent slip. On the lower end stands a Chinese-made wood steam launch being repaired."

The photographs were all taken by Im Shun San, and are very creditable productions, which we take pleasure in presenting to our readers.

* * *

HARDENING LARGE CUTTERS.

The last issue of "Sparks from the Crescent Anvil" gives an illustration of some large cutters of Crescent steel that were hardened successfully in the old-fashioned way of heating and dipping in pure water. The two largest cutters were 8¼ inches in diameter and 10¼ inches long, and there were three others that were somewhat smaller. The process of hardening is thus described:

"The furnace was prepared carefully by putting in a solid bottom of broken coke, about 8 inches deep, so that in rolling the heavy cutters the bottom would not get out of shape and be full of holes.

"The furnace has a regular Tupper grate bar bottom, and is about 30 by 40 inches inside. It is controlled by a damper at the top of the stack—which is the best place to regulate a furnace for heating to harden.

"The reason for providing a deep bed of fuel is to have plenty of heat and to prevent the bottom from getting out of shape and allowing the air to draw up through in spots, which would cause some of the teeth to become hotter than the body of the tools.

"The large cutters were in the furnace at least one hour before they began to change color. In heating large tools of any kind it is always best to give them time to heat slowly, as then they are practically given a second annealing, which allows the grain of the tool to slowly change and prepare itself for the subsequent shock or contraction in chilling.

"These cutters were all chilled in fresh water. They were taken from the furnace at an even, bright red heat and plunged in the water as quickly as possible, and kept moving around as much as was practicable to create a current or commotion in the tank, so as to prevent air or steam bubbles from collecting on the teeth and causing soft spots.

After keeping them in the water bath for a minute or so, the tools were drawn out and quickly tested with a fine file, to see if they were good and hard; then put back in the bath for a short time, and withdrawn while the body was still very warm and placed in a tank of oil and allowed to cool down in the oil.

"The heat in the tools was probably sufficient, when placed in the oil bath, to heat it up to nearly 200 degrees Fahr. They were left in for four or five hours."

* * *

There is always a period in the career of the machinist, when he arrives at that exalted idea of his skill, that no job is too difficult for him to tackle. This is the time when his mouthings about dexterity in the handling of tools, close work and all that sort of thing attracts the attention of the old-timer, and the latter does no want anything better than a chance to pull the callow party off his perch. This he easily does by proposing that the smart one plane or file up a perfect cube. "Pride goeth before a fall" in every instance of this kind. It used to be a stock catch, and probably is yet; but it is a sure cure for conceit, and never fails to bring the lesson home, that the handicraftsmen are scarce that can produce a perfect cube. The apparent simplicity of the understaking is what embitters the failure, and any attempt to explain how it happened only aggravates a bad case. We know, because we tried it a long time ago, when we thought the toughest job would be as clay in our hands, and—well, it was never explained.—Locomotive Engineering.

DIFFERENTIAL LEVERS.

W. H. Sargent.

Archimedes said "Give me a lever long enough and a place to rest it on, and I will move the world." It often happens in designing weighing machinery that in order to obtain the proper "multiplication" or ratio between the long and short arms the designer cries for a "lever long enough and a place to rest it on."

Now, the length of the short arm is limited by the conditions of construction, as will be seen in Fig. 1, since it would evidently not be possible to set the pivots any nearer together than they are shown, and if the other arm is the required number of times longer the whole lever will occupy an inconvenient amount of room. To avoid this, the Fairbanks Scale Co. make use of an ingeni ous arrangement analogous to differential gearing.

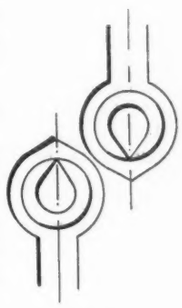


FIG. 1.

Referring to Fig. 2, suppose a load of 25,000 pounds to be suspended from the pivot A. This lever is so proportioned as to transmit one-fifth of this load, or 5,000 lbs., to the pivot B, the remaining four-fifths being transferred to lever C, and is divided so that one-fourth, also 5,000 pounds, comes at the pivot D. Both these levers are suspended from the lever G, and it is evident that if the pivot H is midway between F and E, and the same load, 5,000 pounds, be hung at each of these points, the lever will remain in equilibrium, but if pivot H be moved ever so little out of the center, then one arm becomes longer than the other, and the load on the long end overbalances the same load on the short end and goes down. The poise may now be run

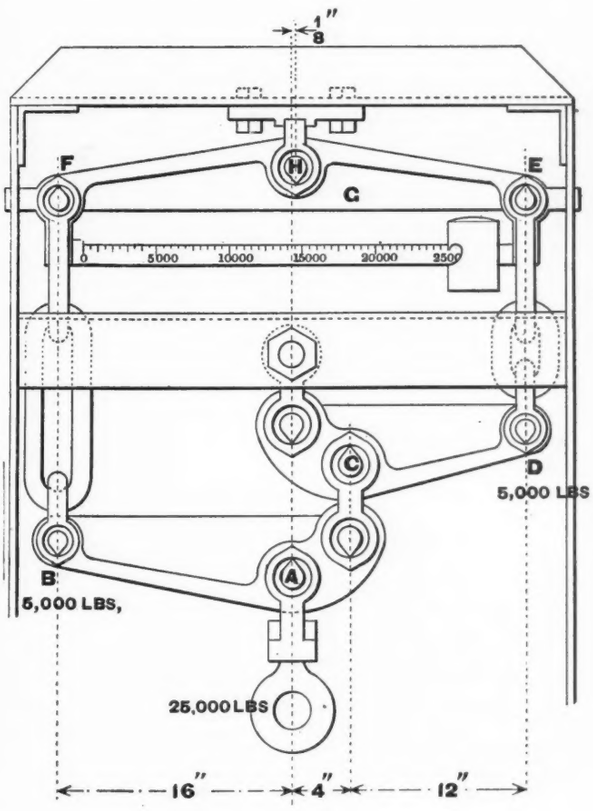


FIG. 2.

out until balance is restored, the reading on the beam at this point indicating the weight of the load. In this way a lever may be obtained "long enough" in effect without being inconveniently long in practice. The sketch shows an arrangement only 30 inches long from pivot to pivot, which is equivalent to a lever having a long arm of 26 feet 5½ inches and a short arm of 1 inch!

F. F. HEMENWAY.

Frank F. Hemenway, the well-known mechanical engineer and former editor of the "American Machinist," died at Brooklyn, N. Y., on Oct. 14, at the age of nearly 61 years. He was born at Hartford, N. Y., Oct. 29, 1837, and served an apprenticeship at the shop of Edward Joslin, Keene, N. H. He received a common school training, but this was only the beginning and the smaller part of his education; for he was by nature a student, and the study of mechanics was more of a pleasure to him than any amusement in which he could indulge. His mechanical education was broad and thorough, and was obtained through his own efforts without the aid of school or teacher.

He was married at Brattleboro, Vt., in 1860, and in 1861 enlisted and served under Gen. N. P. Banks in the civil war. At this time he was employed as a machinist at the Putnam Works, at Fitchburg, Mass., and after his term of enlistment he followed his trade at several points in the West.

The most important part of Mr. Hemenway's work in connection with the construction of machinery was for a period of about ten years at Troy, N. Y., where he was at the Troy and Boston (now the Fitchburg) R. R. shops and later at the works of Starbuck Bros., which were then in a prosperous condition. While at the railroad shops he became intimately associated with the late Geo. W. Richardson in the experimental and other preliminary work connected with the development of the Richardson pop safety valve and Richardson balanced slide valve. During the latter part of his time at Starbuck Bros. he was foreman of the machine shop, and invented the Hemenway automatic cut-off engine. This was a horizontal engine of the four valve type,with a releasing gear driven by cams on a horizontal cam shaft, and receiving their motion from the main shaft through bevel gearing. Several engines were built and operated successfully and with good economy. Their mechanical features were excellent, and were described in the "American Machinist" of March 19, 1881. While in Troy Mr. Hemenway became a charter member of the American Society of Mechanical Engineers.

In 1879 Mr. Hemenway became a frequent contributor to the "American Machinist," and as was natural, in view of his experience at Troy, he wrote mostly upon steam engineering subjects. This was at the time when the steam engine indicator was coming into increasing prominence, and his articles upon this subject attracted widespread attention and subsequently were revised and published in book form, under the title "Indicator Practice and Steam Engine Economy." This book has had a large sale, and is a standard authority to this day.

In July, 1881, Mr. Hemenway became associate edtior of the "American Machinist," and in July, 1887, he became editor, which position he held until October, 1894, when he resigned. Since that time he has been a regular contributor of several leading mechanical papers, and has recently published a second book entitled the "Catechism of the Power Plant," which promises an even larger sale than his former work. At the time of his death he was engaged upon other important literary work. Since February, 1895, he has acted as consulting engineer for MACHINERY, and in this capacity has rendered frequent and valued advice to the editorial staff of the paper. He has always shown a deep interest in the success of MACHINERY, and his assistance will be sorely missed.

Mr. Hemenway's writings have displayed an individuality and strength of character that no one who was at all familiar with them could fail to note. They have borne the stamp of reliability and sound judgment that was but a reflection of the same characteristics in the man. His opinions as a mechanical expert were much sought, and largely, we think because whatever opinion he expressed, either by pen or by word, was so evidently dominated by a common-sense view of things that led him to render correct judgment in nearly every case.

Common sense and kindliness of character. These were the two rare possessions of Mr. Hemenway as we knew him. While his common sense views made his opinion widely sought, his kindly ways made him universally liked. We must add, too, that he displayed these qualities in the face of trials and misfortunes that fell to his lot to a degree, fully appreciated, perhaps, only by those who knew him intimately. He bore up bravely, however, and was ever ready to meet his friends with a cheerful expression or to lend a helping hand wherever he thought it would do good. Many men owe much of their success to the assistance that Mr. Hemenway has so freely given in this way.

CAM CUTTING.

PRACTICAL DATA — THE RESULTS THAT MAY BE OBTAINED WITH A VARIABLE FEED.

WARREN E. WILLIS.

The development of automatic machinery for nearly every purpose has compelled the use of cams to an extent that would not have been considered good practice a few years ago.

The reason is quite evident, for most movements needed by engineers can be readily developed and generally by more than one method. While some motions are to be obtained by cams, others are to be had by levers and links, as used in old-time practice; yet our best designers are choosing cams in preference, regarding them in a deservedly favorable manner, and some data regarding the making may be interesting to those who may have occasion to make use of them.

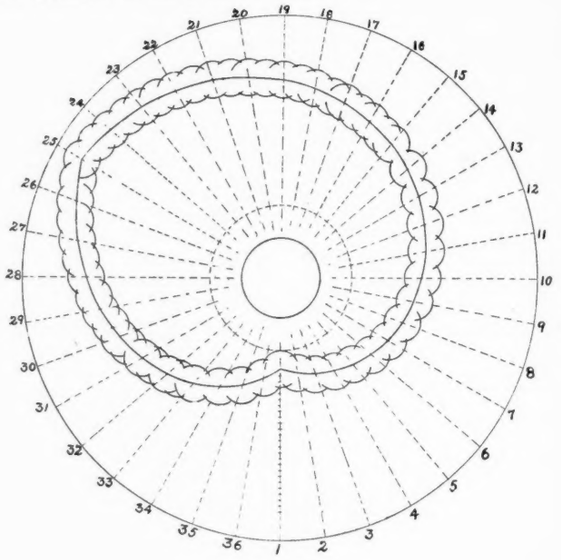


FIG. 1.

The "laying out" of cam curves involves the knowledge of nothing more than is supposed to be known by any designer who has the ability to see their utility.

The name "cam" as used alone may signify any irregular shaped piece, which, by partial or complete rotation, causes other

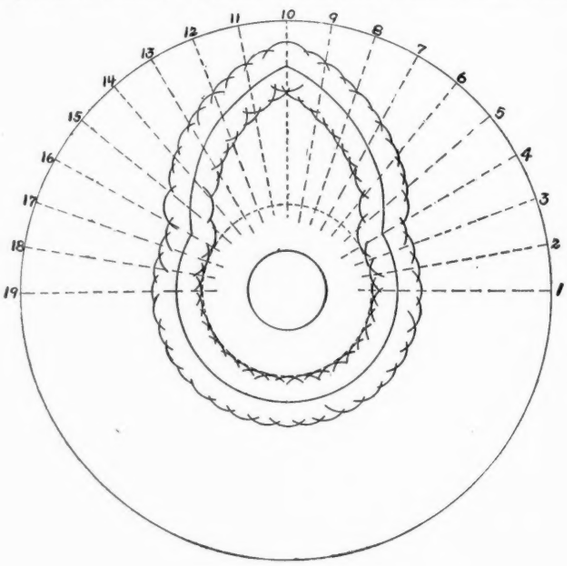


FIG. 2.

parts, suitably attached, to move in certain defined paths; there is no mistaking the meaning of this word, but the terms descriptive of the various forms of cams are not so clear, and it is not uncommon to be mistaken in their meaning.

Mechanics sometimes speak of face and edge cams, but which is which? If the face of a pulley is the periphery, or that part

on which a belt would run, what reason is there for calling the corresponding part of a cam by some other name, and that part that answers to the sides of a pulley, the *face* of the cam?

Axial and radial, scroll and cylinder are common descriptive terms, more or less appropriate to cams cut respectively upon the ends and convex surfaces of cylindrical bodies. There is certainly room for choice.

The usual substance employed is cast iron; after that comes steel, either castings, forgings or bar, and not a few have bodies of cast iron, with steel pieces set in at points where the greatest resistance occurs, the steel pieces being removed, hardened and reinserted after cutting.

Gray cast iron wears well and is extensively used; in fact, ordinarily it is not the cam groove that wears most, but the roller on its outer surface and on its stud, which is reduced in diameter and worn out of round. The action of the roller in its path being constantly turning, there is little sliding friction, and consequently little wear when working properly.

The production of cams accurate to pattern or former, with width of groove true to gauge and smooth as to finish, is a problem that is not so easy of solution with the majority of cam cutting contrivances in common use. Most of these contrivances are in the nature of attachments to other machines, usually milling machines or lathes, which not being made with this particular object in view, too often fail to produce satisfactory work.

A positively rigid work holding device, supported and carried by suitable guides, absolutely controlled by some kind of

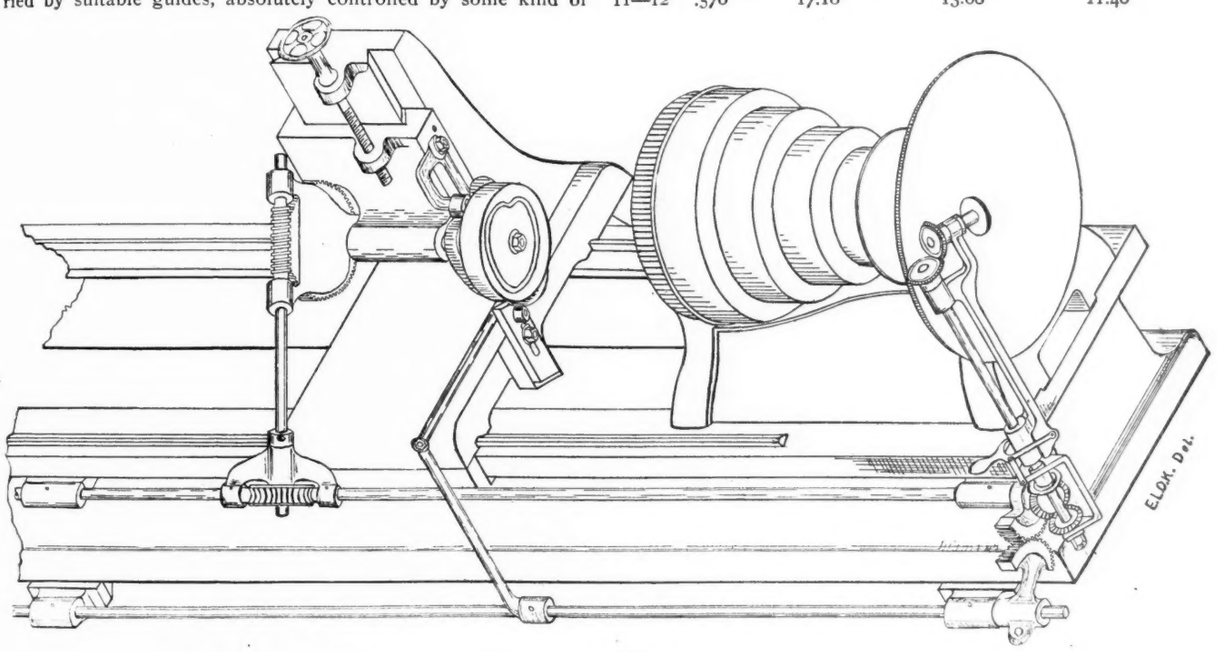


FIG. 3. CAM CUTTING DEVICE.

former, and an equally stiff cutting tool, are fundamental necessities. The number of cuts may vary according to the nature and quantity of stock to be removed, and also as to the nicety of the required finish.

In most cam cutters the speed of the cutter is uniform and proportional to the diameter of the cut and the hardness of the stock; the work also revolves at a uniform speed, and here is where exception is taken to the methods in vogue, for if the work revolves at a regular rate of speed during the entire time of the cut and the speed of its revolution is for a certain number of feet per minute, it will be right only in that part of the groove having such a radius that its circumference multiplied by its rate of revolution would give that number of feet travel per minute. As the groove approaches the center, the feed becomes too slow, and as it moves away from the center it would become dangerously fast were it not for the fact the machine is speeded so that when the cutter is at the outside of the cam the feed is not beyond the safe limit. As only a part of the cut is taken at the outer edge, however, this arrangement shows a waste of time. A parallel case is noticed in facing up, or squaring off, work in a lathe, and everybody recognizes the necessity of slowing down the speed as the tool moves outward. Another point in producing fine work is that in turning corners of more or less abruptness, the cutting speed should be retarded to correspond with the above reasoning.

It will, of course, be understood that these remarks refer to a scroll cam groove, cut radially upon the end surface of a cylinder. Two cases in point are here given that show the saving that can ordinarily be effected by proportioning the feed to the actual cutting length of the cam groove.

Tables Showing Time by Proportional Feed Method.

Heart Cam (Fig. 1.)

Diameter of blank	11.000 inches
Diameter of hole	1.625 "
Diameter of cutter	.875 "
Longest radius of throw	4.875 "
Shortest radius of throw	1.875 "
Speed of cutter	160 revs.
No. of divisions on cam face	36
No. of degrees to division	10
No. of degrees in outward movement	240
No. of degrees in inward movement	120

No. of space.	Length of cut.	Rate of feed per minute. 2 inches.	2.5 inches.	3 inches.
1— 2	.370 inch	11.10 seconds	8.87 seconds	7.40 seconds
2— 3	.375 "	11.25 "	8.99 "	7.50 "
3— 4	.385 "	11.55 "	9.23 "	7.70 "
4— 5	.400 "	12.00 "	9.59 "	8.00 "
5— 6	.420 "	12.60 "	10.07 "	8.40 "
6— 7	.445 "	13.35 "	10.66 "	8.90 "
7— 8	.470 "	14.10 "	11.26 "	9.40 "
8— 9	.495 "	14.85 "	11.77 "	9.90 "
9—10	.520 "	15.60 "	12.46 "	10.40 "
10—11	.545 "	16.35 "	13.07 "	10.90 "
11—12	.570 "	17.10 "	13.66 "	11.40 "
12—13	.590 "	17.70 "	14.14 "	11.80 "
13—14	.610 "	18.30 "	14.62 "	12.20 "
14—15	.630 "	18.90 "	15.10 "	12.60 "
15—16	.650 "	19.50 "	15.59 "	13.00 "
16—17	.670 "	20.10 "	16.06 "	13.40 "
17—18	.690 "	20.70 "	16.54 "	13.80 "
18—19	.710 "	21.30 "	17.02 "	14.20 "
19—20	.735 "	22.05 "	17.62 "	14.70 "
20—21	.760 "	22.80 "	18.22 "	15.20 "
21—22	.785 "	23.55 "	18.82 "	15.70 "
22—23	.810 "	24.30 "	19.42 "	16.20 "
23—24	.835 "	25.05 "	20.02 "	16.70 "
24—25	.860 "	25.80 "	20.62 "	17.20 "
25—26	.885 "	26.55 "	21.22 "	17.70 "
26—27	.840 "	25.20 "	20.14 "	16.80 "
27—28	.795 "	23.85 "	19.06 "	15.90 "
28—29	.750 "	22.50 "	17.98 "	15.00 "
29—30	.705 "	21.15 "	16.90 "	14.10 "
30—31	.660 "	19.80 "	15.80 "	13.20 "
31—32	.615 "	18.45 "	14.74 "	12.30 "
32—33	.570 "	17.10 "	13.66 "	11.40 "
33—34	.530 "	15.90 "	12.71 "	10.60 "
34—35	.490 "	14.70 "	11.75 "	9.80 "
35—36	.450 "	13.50 "	10.79 "	9.00 "
36— 1	.410 "	12.30 "	9.83 "	8.20 "
Total time in secs.		660.80 "	528.00 "	440.80 "
Total time in mins.		11.01 "	8.80 "	7.34 "

Lifting Cam (Fig. 2.)

Diameter of blank	9.5 inches
Diameter of hole	1.375 "
Diameter of cutter	.875 "
Speed of cutter	160 revs.
Longest radius of throw	4 inches
Shortest radius of throw	2 "
No of divisions on cam face	36
Part of cam in action	120 degrees

No. of space.	Length of cut.	Rate of feed per minute. 2 inches.	2.5 inches.	3 inches.
1— 2	.35 inch	10.5 seconds	8.4 seconds	7. seconds
2— 3	.35 "	10.5 "	8.4 "	7. "
3— 4	.35 "	10.5 "	8.4 "	7. "
4— 5	.48 "	14.4 "	11.5 "	9.6 "
5— 6	.52 "	15.6 "	12.4 "	10.4 "
6— 7	.57 "	17.1 "	13.6 "	11.4 "
7— 8	.63 "	18.9 "	15.1 "	12.6 "
8— 9	.69 "	20.7 "	16.5 "	13.8 "
9—10	.75 "	22.5 "	18.0 "	15.0 "
1—10 (tot.)	"	140.7 "	112.3 "	93.8 "
10—19 (tot.)	"	140.7 "	112.3 "	93.8 "
19— 1	6.30 "	189.0 "	151.3 "	120.2 "
Total seconds		470.4	375.9	308.2
Total minutes		7.840	6.265	5.136

As the rise and fall of the cam grooves are alike, it is only necessary to compute one side, or one-half the action, and as the side opposite that having the action, or throw, is of a uniform radius each section will be alike therein.

The figures, as originally drawn, were approximately and practically correct, and were scaled to obtain the above results. As here reproduced, however, they are shown at a considerably smaller scale.

The total time ordinarily taken may be arrived at by considering the longest radius as the controlling feature, and calculating from it direct or by finding the length of the longest division from a drawing or otherwise.

The results both ways are here shown:

Tables Showing Time Ordinarily Taken.

Lifting Cam.

Feed per minute.	2 inches, min.	2.5 inches, min.	3 inches, min.
Time by direct measurements	13.500	10.800	9.000
Time by using long radius	12.566	10.053	8.377
Difference	.934	.747	.623

Heart Cam.

Feed per minute.	2 inches, min.	2.5 inches, min.	3 inches, min.
Time by direct measurements	15.93	12.73	10.62
Time by using long radius	15.32	12.25	10.21
Difference	.61	.48	.41

These differences are due to the fact that the length of each division of ten degrees in a circle, having a radius of 4 inches, is .698 inch, while in the lifting cam the longest distance in any division is .75 inch, the additional length being due to the curve of the groove.

In the case of the heart cam, the length of each division on its radius of 4.875 inches is .85055 inch, while the actual measured distance of cam groove is .885 inch, the next longest being .860 inch, and all the others less than that; it is therefore assumed that the radial speed is safe, and, being more convenient, is the one usually taken.

This being the case, the saving in time is as below, and it may be said that experience has proved it to be correct.

Tables Showing Saving by Proportional Feed.

Lifting Cam.

Feed per minute.	2 inches, min.	2.5 inches, min.	3 inches, min.
Time by common rev. feed	12.566	10.053	8.377
Time by proportional feed	7.840	6.265	5.136
Gain in time effected	4.726	3.788	3.241

Heart Cam.

Feed per minute.	min.	min.	min.
	2 inches,	2.5 inches,	3 inches,
Time by common rev. feed	15.32	12.25	10.21
Time by proportional feed	11.01	8.80	7.34
Gain in time effected	4.31	3.45	2.87

Cam Speed Controlling Device.

The drawing (Fig. 3) shows the cam speed controlling device used for this work as it was attached to a cam cutter consisting primarily of an ordinary engine lathe bed and head. The usual carriage is replaced with a special attachment, having an apron in front carrying gears connected to the rack for moving it along the shears as usual. It also is provided with slides for vertical action of the part which forms a bearing for the work spindle; the said vertical slide can be raised by the screw, as shown, or the screw may be turned so as not to affect the action of the former, or master cam, that is directly in the rear of the blank to be operated upon.

On the rear end of the work spindle is a worm gear, driven by the worm shaft, as shown, which in turn receives its motion from a worm gear at its lower end, driven direct from the upper or feed shaft attached to the rear of the lathe; it is evident as the slide is raised or lowered that this shaft will alter its angle, pivoting on the shaft which drives it.

Now, this feed shaft is usually driven by means of small cone pulleys at the end, belting from the lathe spindle feed cones; in this case the cones are removed and in their place is a leather covered friction disk, driving a wheel attached by gearing to an upright shaft; on this also are two gears and means for throwing either or neither out of or into connection with the feed shaft proper, thus providing means for stopping or reversing the feed at will.

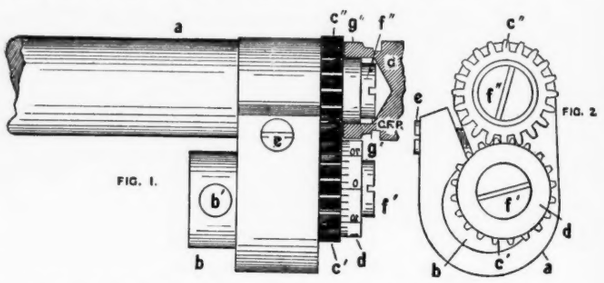
It will now be evident that the relative speed of this feed shaft is dependent on the position of the small wheel on the friction disk, and means for controlling and varying this position are provided for by the two gear segments, one of which is rigidly attached to the framework supporting and carrying the gears and light shafts, while the other is adjustable on a secondary shaft, below the feed shaft. This is splined for a collar attached to a system of bell crank levers and acted upon by a roll in connection with the master cam, the revolution of which not only acts to produce the desired shape by raising or lowering the cutting point, but at the same time effects such changes in the speed of the feed as to give an equal and uniform cutting rate.

* * *

INDEXING SMALL MICROMETER COLLARS.

I recently designed a set of tools for making a quantity of small micrometer index collars for the cross-feed and compound rest ball crank. I made them from the solid bar in a screw machine, and indexed them before cutting off by the accompanying illustrated device.

G represents the end of the bar with a collar, finished and ready to be cut off at *g'*. The face *g"* is to be indexed.



A forms the body of the tool, with a shank to fit into the turret; *b* is an eccentric carrying gear, *c'* and knurl *d* keyed together and revolving freely on screw or stud *f'*. Knurl *d* is the reverse of the desired indexing with raised lines and figures.

The hub of gear c" is made to fit tightly into the work, so as to form a positive drive to the knurl and insure accurate results.

The knurl is moved to and from the work by a pin lever inserted in hole *b'* of eccentric *b*. Screw *e* is designed for adjustment and to give friction enough to eccentric *b* to keep it in any position desired, while the other operations of the machine are executed.

The diameter of *g"* must be the same as knurl *d* and the pitch line of the gears *c'* and *c"*.

C. F. P.

* * *

Prof. D. L. Jackson, of the University of Wisconsin, has been collecting data which show that one student in every 10,000 inhabitants is taking an engineering course of high grade.

ELECTRICALLY DRIVEN MACHINERY.

WHERE MOTORS CAN BE USED FOR INDEPENDENT DRIVING—CONSTANT SPEED AND VARIABLE SPEED MOTORS—REVERSING THE MOTION —POINTS ABOUT STARTING AND STOPPING.

WILLIAM BAXTER, JR.

Several years ago some of the enthusiastic admirers of electric motors began to urge that they be used to operate machinery by being directly connected thereto, thus avoiding the use of belting. By many this innovation was opposed, on the ground that the trouble of taking care of as many motors as there were machines would render the plan impracticable, and considering the uncertainty of action of motors in those days, their arguments were not wholly groundless, although sometimes rather one-sided. The advocates of the electric system were equally biased in their way, but the result of the contention was, that more impartial minds took up the question, and, as we know, direct connected machines are very numerous, and in most of the cases where such combinations have been made, it has been found that the new arrangement is an improvement on the old style. There are many who are inclined to assume, from this, that under any conditions direct driven machines are the best, but such is not the case. In order to be able to determine whether direct driving is the best or not, in any particular case, it is necessary to weigh the advantages and disadvantages, so as to see which predominates. The advantages of direct driving are that the power consumed is less, because the motors only run when the machine runs; that belting is dispensed with, and thus dangers incident thereto are removed; that the atmosphere is less vitiated, the whole premises kept cleaner, the light less obstructed and finally that there is more room overhead, and hence work can be handled with cranes with greater freedom. As an offset to this may be stated that motors do not give back all the power they consume, the loss ranging from 10 to 15 per cent.; that a number of small motors may cost more than the shafting and belting they replace, and that the cost of keeping them in repair will be greater.

Assuming that motors are the more expensive in first cost, it follows that they cannot be used profitably unless there is a reduction in the running expenses effected thereby. Whether such a reduction can be made will depend upon the nature of the machinery to be operated. If it is of the kind that is kept in constant or nearly constant use, it is evident that there is very little room for a saving in power, and what little may be gained in this way may be balanced by the loss in the motors and generators. In establishments where the machinery is not in constant use, as, for example, machine shops, printing establishments, etc., a considerable saving in power can be effected by the direct application of motors to the individual machines. Tests that have been made show that in some shops of this kind, the use of the machinery is so intermittent that more than one-half of the power is used to keep the shafting in motion, and in some extreme cases as much as four-fifths. In many cases, where nothing could be gained by direct connection, it still might be found advantageous to run each line of shafting by a separate motor, but it is not the purpose of this article to enter into a consideration of that phase of the subject. So far as the direct application of motors to individual machines is concerned, it may be assumed as very nearly certain that nothing can be gained in the way of economy of operation unless the machines are of the class that under practical conditions cannot be kept in constant operation. Having settled upon this point, the next question is as to the kind of motor to use in any particular case. Motors are made to run at a constant speed without any regard as to how the load changes, and they are made so as to vary the velocity with changes in load or at the will of the operator Machinery is also made so that it must run either at a constant velocity or be under the control of the operator, hence the type of motor will depend upon the kind of machine it is to run. Generally speaking, the constant speed motor is the best suited for any case, because, as a rule, where a variable velocity is required, the range of speeds is more than can be given advantageously by the motor itself, and better results are obtained when the changes are effected by means of gearing. To illustrate this point, suppose it is desired to run a lathe, the speed of which may be a few revolutions per minute when the backgear is thrown in, and a thousand or more when the belt is on the smallest step of the cone. If an attempt were made to simplify the machine by removing the cone and back gear and attaching a motor directly to the main spindle, the results would not be satisfactory, because the low velocity would be entirely too low for the motor; it could not run except at a great loss of power, and would not only be expensive, but in all probability fail to develop sufficient force to keep the lathe in motion at the slow speeds, if taking a heavy cut. If the lathe is left intact, with all its gearing, and the motor is attached to it so as to drive the cone in the same way as when driven by a belt, the results will be of an entirely different character. Under these conditions the motor will be able to run always at the same velocity, and thus work to the best advantage, and, in making changes in the speed of the lathe, there will be no danger of injuring the motor, or stalling it when taking heavy cuts.

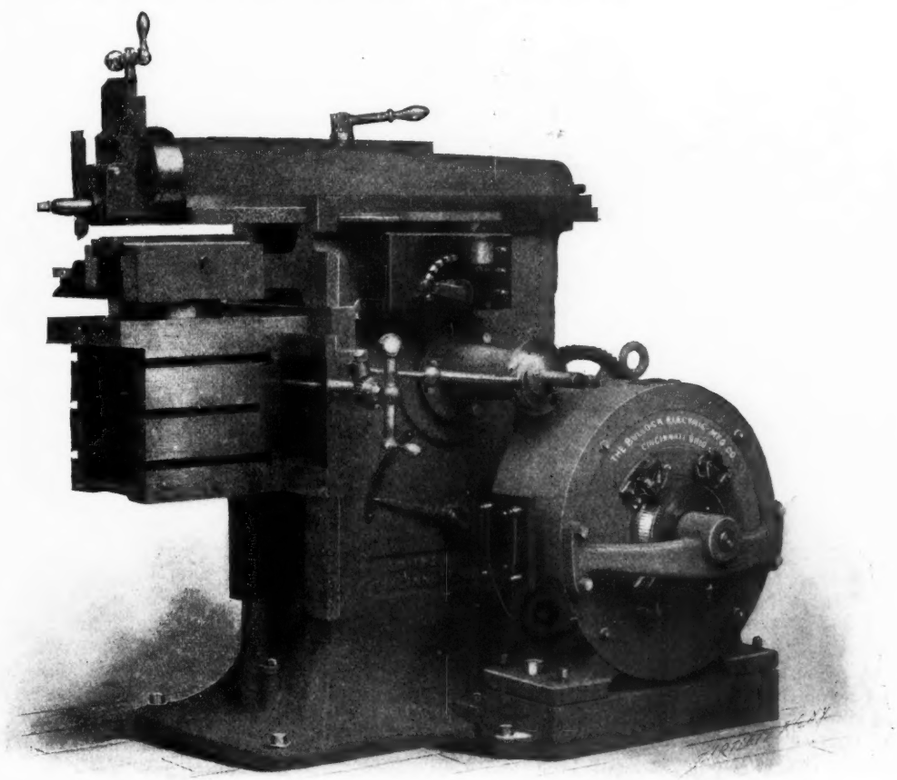


FIG. 1. MOTOR OF THE IRON CLAD TYPE.

If a machine is of such a character that it is necessary to reverse its motion often, and with the least possible loss of time, a direct connected motor will not give satisfactory results, because its direction of rotation cannot be reversed instantly. On this account it can be seen that such an arrangement would not work well on a lathe, unless the motor were attached to the

countershaft or to a shaft connected with the driving cone by means of two sets of gearing, so that backward and forward motion could be obtained by throwing a clutch, and without reversing the motor. There are many machines, however, in which reversing the motion is not necessary, and with these the direct connection becomes much more simple. Among such machines may be mentioned shapers, drill presses, printing presses, etc.

Almost every form of machine that can be advantageously driven direct connected can be found in the market at the present time; but when it is desired to extend the method to some machine not now made, it can be determined whether the result will be satisfactory or not from the nature of the machine. If it is to run in both directions, and does not contain within itself the necessary reversing gears, it cannot be driven

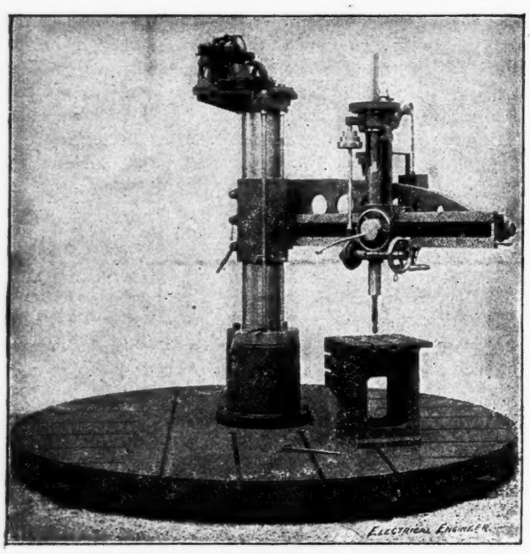


FIG. 2. ARRANGEMENT OF MOTOR FOR RADIAL DRILL.

direct with good results. If it is to run at widely differing speeds it must have reducing gearing, so that the motor may be run at a constant velocity. If the direction of motion is reversed occasionally, and it is not necessary that the reversal be effected almost instantly, the motor can be connected direct and will work well. If the speed change is small, the motor may also be direct connected. In the latter case a variable speed motor will give the best results. In cases where it is desired that the speed remain as nearly uniform as possible, variable speed motors will not answer, even if the changes in velocity are small, because these machines have no means for maintaining a constant velocity, and the result is that when the load is increased, they will run slower, and when it is reduced they will run faster. The only way in which the speed of such motors can be controlled is by the movement of the regulating switch. If the power required to operate the machine is nearly constant, as is likely to be the case with printing presses, a variable speed motor will answer the purpose.

The difference between constant speed and variable speed motors is wholly in the winding of the field coils; the former are what is known as shunt wound, and the latter as series wound. Sometimes the constant speed motors are differentially wound. If a motor is shunt wound the field wire will be very small, and if it is series wound the wire will be about double the cross section of the armature wire. If the field is differentially wound, it may not be easy to determine from the outside appearance whether it is series or differential, because in the latter winding the field has both fine and coarse wire, and the coarse is generally placed on the outside. In some cases the coarse wire is wound under the fine, and then it is difficult to determine whether the machine is shunt or differentially wound. In any case, however, the character of the winding can be ascertained by noticing the number of ends of wire that come from each coil. If the winding is either plain, shunt or series, there will be only two ends coming from each coil, as there is only one length of wire; but if the winding is differential, there will be four ends, as there are two separate wires wound on each coil. The type of winding can always be ascertained by running the motor, for if it is shunt or differential, it will not run away when started up without a load, but if it is series wound it will attain a very high velocity under the same conditions, so high that any one familiar with machinery will be able to see at once that it is turning at a velocity far beyond the safe limit.

In addition to determining the type of motor to use for any particular case to be able to impart the proper motion to the machine, it is also necessary to decide from the character of work the machine performs, whether it is advisable to attach the motor to it or not. There may be an objection to the direct attachment on account of liability of injury to the motor or the product of the machine. As to the motor, it should be kept in mind that although, if well made, it can withstand the destructive effects of oil, metallic dust and dirt of any kind for a considerable period of time, it will eventually come to grief. The length of time it will run under such conditions is entirely uncertain, and may be a day or several years, but prudence dictates that we act upon the assumption that it will give way in the shorter period. In view of this fact, a motor should not be attached directly to any machine where there is danger of its being covered with dirt, unless a place can be found where it will be well shielded. From this it might be inferred that direct connection with metal working tools would not be successful, but as the chips thrown out by such machines are generally confined to one spot, there is no difficulty in locating the motor where it will be beyond their reach, as is shown in Fig. 2, where the motor is mounted at the top of the tool.

Many of the motors made at the present time for direct connection are wholly inclosed in an iron shell, and are known as the ironclad type. A motor of this kind is shown in Fig. 1. As such machines are better protected than the open designs, they can be located in positions where the latter would not be safe.

So far as the machine to be driven is concerned, about the only thing to consider is whether there is anything about it that can easily take fire. If such is the case a motor should not be attached to it unless proper safeguards are provided to prevent setting fire to it or its product in case of disarrangement of the vital parts of the motor. A properly designed motor, if in good adjustment and clean, is not at all liable to set fire to even decidedly inflammable material; but if something goes wrong with it, the reverse is the case. Ordinarily the only place in a motor where a spark can be seen is at the commutator, and this will be so small generally as to hardly attract attention, but if a brush gets out of adjustment, the spark can change to a veritable blaze. This may also be said of any of the connections. All of these are perfectly harmless when tight, but when they become loose they can do considerable damage. Generally speaking, the danger arising from these causes is so small as to be hardly worthy of notice, but it is nevertheless a remote possibility, and if it can be guarded against it is best to do so.

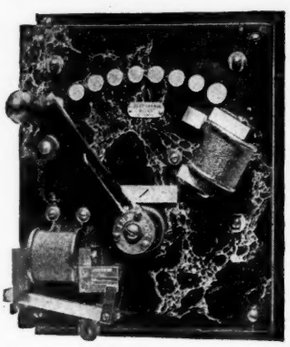


FIG. 3. SWITCH FOR CONSTANT SPEED MOTOR.

If the commutator is covered, as is the case in some motors, no damage can be done outside of the motor should the brushes become disarranged. On this account machines of this type are better adapted to cases where the machine driven or its product are of an inflammable character. If the motor is not of the protected type, it can be rendered equally safe by being encased with sheet iron at the commutator ends.

To start a motor it is necessary to have a suitable switch, just as in the case of a steam engine it is necessary to have a throttle valve. The throttle valve is generally placed very close to the engine, and never at any great distance, but with motors the starting switch may be near by or at any desired distance. The kind of switch required depends upon the type of motor. Fig. 3 shows one of the many forms used in connection with consant speed motors, and Fig. 4 one of types used with variable speed motors.

The switch shown in Fig. 3 is connected with a resistance, each one of the circular contacts in the path of the handle being connected with a different point in the resistance. When the handle is moved to cover the first contact, the current passes through the motor, and that portion of it that goes through the armature also passes through the resistance with which the starting switch is connected. The object of introducing this resistance in the armature circuit at the start is to prevent the current from rising to such a strength as to burn out the armature, and also to cause a sufficient amount to pass through the field coils to give the motor a strong turning force, so that it may be able to set the machinery it drives in motion as rapidly as possible. As soon as the motor begins to get under headway, the operator moves the switch handle over the circular contacts, one at a time, until the last one is reached, by which time the motor will be running at its normal velocity. The time consumed in moving the switch over to the last point will depend upon the effort the motor has to make in getting under headway. If it is running with a light load, it will speed up very fast, and the handle can then be moved around equally fast, but if the load is heavy the speed will not increase so fast, and then it will be necessary to move the handle slower. Generally the time required should not be over five or ten seconds. A man who has had even a small amount of experience in starting a motor can judge from the way in which it picks up speed just about how fast to move the handle. The only precaution to be observed in operating these switches is to not allow the lever to lap over the end of a contact so far as to just touch the one ahead of it, as if this occurs, a spark will jump to the forward contact and roughen it up, and in time the result will be that the contacts will become so rough and worn away as to require truing up, or at least polishing. In stopping a motor the switch must be moved as quickly as possible; if not, the contacts will be burned unnecessarily. As the handle passes from one contact to another in stopping, there will be a spark, as each contact introduces additional resistance into the circuit. The slower the switch is moved the longer the sparks will keep up, therefore the more they will burn the edges of the contacts.

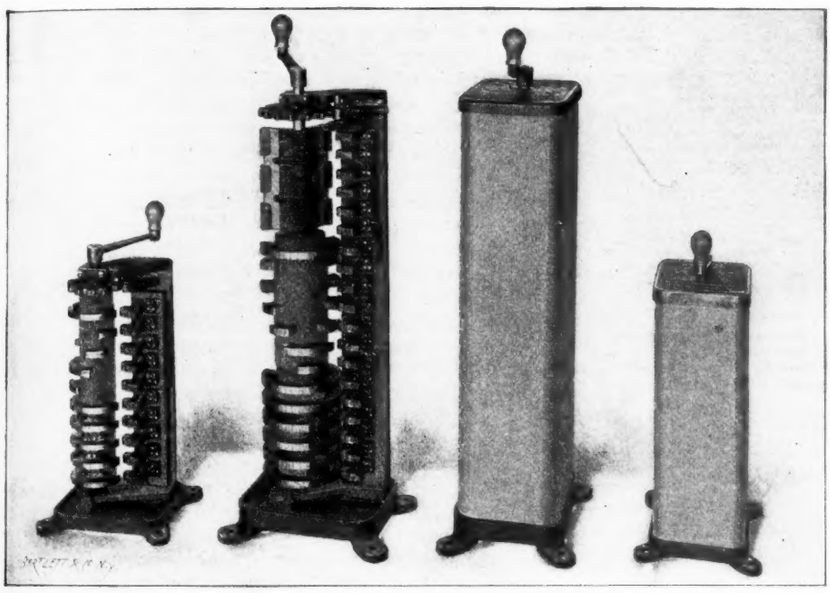


FIG. 4. SWITCHES FOR VARIABLE SPEED MOTORS.

Many of these starting switches are made so that if the motor is loaded to such an extent that the current is liable to overheat the armature, they will fly open. These are called safety, or automatic, starting switches, and are the only proper kind to use. If when a motor is running the current should stop, which is possible through some momentary disarrangement at the generating station, the motor will stop, and if then the current is turned on again, damage may be done, because there will be no resistance except that of the motor armature, which is very low, to hold the current back, and there will be an enormous current passing through it. To prevent injury in such cases it is necessary to provide some means of breaking the current before it can do damage. The most simple arrangement is to introduce into the circuit a short length of wire made of some metal that melts at a low temperature, as lead. This should be of such size that it will melt with a current slightly larger than the motor can carry safely, and if it is so proportioned, it will fuse, and thus break the circuit before any damage is done. Wire of this kind is made and can be had at any store that deals in electrical supplies. It is known as fuse wire, and is made of such diameters as to melt with currents of different strengths, so that the proper size for any case can readily be obtained. These fuse wires cannot act very accurately, or, more properly speaking, they cannot be used of such size as to melt with a current only slightly in excess of the strongest the motor can carry, for if they were adjusted so closely, the result would be that every time the load went beyond a certain point the fuse would give away and the motor would stop. This condition of things would be very annoying, and to prevent it it is customary to use fuses that will melt with a current about 50 per cent. stronger than that required to give the full power of the motor. The mistake should never be made, however, of using a fuse entirely too large, for then the motor will have no protection at all.

Some starting switches are so made that if the current stops they will fly open automatically and thus protect the motor. These are the best kind to use. Beside safety fuses there are magnetic circuit breakers, that open the circuit by means of a switch when the circuit become dangerously strong. Circuit breakers of this kind should always be used in connection with large motors.

The switch described in the foregoing is only suitable for use in connection with constant speed motors. For variable speed machines the type shown in Fig. 4 is used. As will be seen, this is far more substantial and complicated than the one just described It must be more substantial because more frequently used. In a constant speed motor the current divides into two parts, one passing through the armature and the other through the field coils. The latter is small compared with the armature current, and ranges from about one-tenth in small motors to about one-hundredth in the largest sizes. In variable speed motors the whole current passes through both field coils and armature. The speed is varied by introducing resistance into the circuit, so as to cut down the strength of the current, or by increasing or decreasing the number of turns of wire in the field coils so as to reduce the strength of the magnetism. If the regulation is effected by means of resistance, this has to be much larger—that is, of great current-carrying capacity—than that used with constant speed motors, because it is in use nearly all the time while with constant speed machines it is only used in starting. The switches are made so that when the switch lever or its equivalent passes from one contact to another, a pawl drops into a notch, this being done so as to prevent stopping the handle when only partly on the contacts. To increase the speed the handle is turned around so as to cut out the resistance, but if the speed is too great, it is necessary to swing the handle around to the starting point, and then move it forward again. If this is not done the contacts will be badly burned. All the contacts in these switches are made very heavy, so as to stand rough usage, but for all that they will give out if neglected. Whenever the surface becomes rough or burned, they should be polished off with emery cloth. As variable speed motors are controlled wholly by the movement of the switch, smooth running depends upon the skill of the operator.

* * *

It will pay to use an attachment on the emery wheel that can conveniently be used at frequent intervals for truing it up.

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NOVEMBER, 1898.

THE AMERICAN BEAM ENGINE.

The London "Engineer" is doing a commendable piece of work in publishing a series of articles upon the beam engine used on American paddle-wheel steamers. This type of engine is distinctively American, and holds even a more important place in the development of steam engineering in this country than does the Cornish engine in the history of steam engineering in England. It deserves a comprehensive and thorough treatment, both from an historical and an engineering standpoint, and this will undoubtedly be accorded it in the columns of the "Engineer."

This form of engine was designed to meet the requirements of the large and luxurious steamers that have plied the rivers and inland seas of this continent during the latter half of the century, and has fulfilled its mission so well that it has almost held its own to this day, in spite of the increasing use of the screw propeller on boats of this class and of high-pressure steam, which makes some other type of engine a necessity.

On paddle-wheel steamers the rotative speed must, of course, be low, and one advantage of the beam engine is that a long stroke of the piston, and consequent high piston speed, can be secured at a low rotative speed of the shaft without monopolizing too large an amount of boat room for the propelling machinery.

The several long connections between the crank and piston also give a degree of flexibility to this type of engine that is a great advantage on shallow boats, like the river steamers that are continually distorted by the heavy loads which they carry. These two features are probably responsible for the continued favor accorded the beam engine for paddle-wheel boats.

Another feature that should be remarked upon is the excellent reputation that these engines have always had for the economical use of steam. The earlier boats carried only about 30 or 35 pounds boiler pressure, but gave an economy that was much better than would be expected were we to judge solely by the pressure of the steam used. Haswell, for example, quotes the "Daniel Drew" and "City of Boston" as using, respectively, 2.2 and 2.1 pounds of coal per horse-power per hour, which is better than the average results of the English Cornish engines of that period. These two boats each carried 35 pounds boiler pressure, and their good record can be accounted for only on the ground that the cylinders were of very large size, and the cylinder condensation correspondingly small. The redeeming feature of these early boats, so far as the economical use of steam is concerned, was that they were generally fast, of good size and consequently required large engine power. With low pressure steam and slow rotative speed large engine power means a large engine cylinder, which in turn means that the area of cylinder wall surface was small compared with the volume of steam used, a condition favorable to economy.

Engineers in this country will follow the series in the "Engineer" with much interest, and we doubt not with much profit.

* * *

STANDARD BUSHINGS FOR JIGS.

It is, perhaps, a little strange that the attempt is so seldom made to systematize jig making. The tendency of to-day is to standardize everything about the shop, from the catalogues and office stationery to the bolts and pins in the stock room. Jigs are used to the greatest extent in shops where standards are adopted, and where the work is done on the interchangeable plan. Nevertheless, the jigs, which are produced for the very purpose of systematizing the work of building the machines, are generally designed and made to suit the taste of the particular man who is making them, according to no particular system.

It is very evident that jig making cannot be systematized to anything like the extent that is possible in making the duplicate parts of a machine. It is one of the first principles that each jig must be as complete in itself as possible to avoid delay, and the liability of a mistake through securing the wrong parts for the piece to be operated upon. In general, also, it is not advisible to have a jig of the ordinary construction adapted to more than one piece, or certainly to more than a few pieces, unless it is arranged so that there is no opportunity for error.

There seems to be no good reason, however, why, in preparing to make a number of jigs, a complete system of interchangeable bushings should not be adopted. As jigs are used largely for locating and sizing holes, the number of bushings in a collection of jigs is inevitably large and represents a considerable sum of money. There are many places where bushings can interchange between the different holes in one jig or in different jigs, and by stamping the sizes of bushings required on each jig and making the outside diameters of the bushings different when they have different inside diameters there would be small chance for a mistake. Where this was done bushings could be kept in the tool room subject to order, and could be examined and tested when returned, just as gauges and reamers should be tested before they are put back on the shelves.

Even though it were not advisable to have bushings interchange, or it should be necessary to have them driven permanently in place, it still would be of advantage to have them conform to a standard as far as the design of the jig would allow. Repairs could be made easier, and the bushings, being produced in quantities, would be much cheaper. Several sets of bushings of different styles would, of course, be required. Some would have to be large enough on the outside to be a drive fit, some would have to be a loose fit, and others would have to vary in their inside diameters according as they were intended for rough drilling, machine reaming or other operation.

Even with a system of standard bushings it would probably be necessary to make many special ones, just as with a system of standard pins and screws carried in stock it is occasionally necessary to make special pins and screws; but we believe that in most cases it would be possible to make use of the standard ones. If any toolmakers or designers have adopted standard bushings for jigs, we should like to hear from them regarding the matter.

* * *

It is a remarkable fact that Boston's subway was built and put in running order at a price below the estimated cost. Where and when did such a thing happen before?

FORMULAS FOR THE STRENGTH OF GEAR TEETH.—2.

J. J. FLATHER.

If in the first case the pitch had been determined from formula (10) the tooth would have only about three-fourths of its assumed strength for a given load P, whereas in the second case the formula would give a tooth about a third stronger than necessary for the same load P.

Since the pitch varies as the square root of the pressure, it will be evident that if we multiply the pitch in each case by the inverse ratio of $\left(\frac{t'}{t}\right)$ we shall obtain a proper measure of the strength of the two teeth, since

$$\frac{1.52}{1.34} = 1.14, \text{ and } \frac{1.52}{1.76} = 0.86,$$

the respective pitches will be 1.14 p' and 0.86 p', in which

$$p' = \sqrt{\frac{15\,P}{f\,x}}$$

from formula (10).

It is evident that the method of laying out the gear tooth will have some influence upon its strength; this is very clearly shown

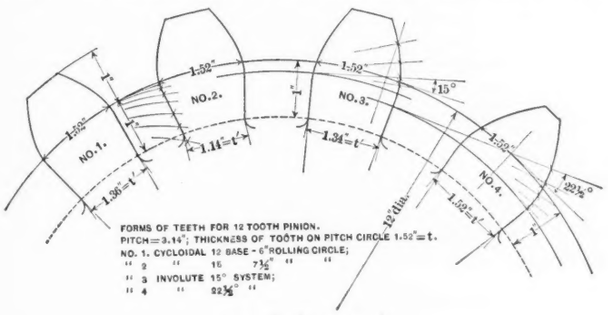


FIG. 8. (REDUCED).

in Fig. 8, which represents a twelve-tooth pinion of 3.14 inches pitch accurately drawn for the four systems represented, viz.: (1) cycloidal, 12 base, *i. e.*, the smallest pair of gears is assumed to have twelve teeth, and these have radial flanks, so that the diameter of rolling circle is one-half of the diameter of pitch circle of a twelve-tooth pinion; (2) cycloidal, 15 base; (3) involute, 15 degrees line of action, and (4) involute, 22½ degrees line of action.

By laying out a series of equidistant gears, according to the different systems represented, the exact thickness of t' and t may be determined graphically, from which the values of the ratios $\left(\frac{t'}{t}\right)$ and $\left(\frac{t'}{t}\right)^2$ may be readily ascertained.

This has been done, and the results are given in Tables I. and II., in which the ratio is presented in the form of a coefficient corresponding to the number of teeth in the gear.

TABLE NO. I. VALUES OF COEFFICIENT $C_p = \frac{t}{t^1}$.

No. of Teeth in Gear.		Value of Coefficient $C_p = \frac{t}{t^1}$.			
		Involute System.		Cycloidal System.	
Exact.	Intervals.	15°.	22½°.	12 Tooth Base	15 Tooth Base
12	12	1.14	1.00	1.12	1.33
13½	13— 14	1.10	.96	1.08	1.19
15½	15— 16	1.07	.92	1.04	1.10
17½	17— 18	1.04	.88	1.00	1.03
20	19— 21	1.01	.84	.96	.98
23	22— 24	.98	.80	.92	.95
27	25— 29	.95	.76	.88	.92
33	30— 36	.92	.73	.85	.89
42	37— 48	.88	.70	.82	.86
58	49— 72	.84	.68	.79	.83
97	73—144	.80	.66	.76	.80
290	145—Rack	.76	.65	.73	.77

Combining these coefficients with formulas (9) and (10) we obtain

$$P = 0.066\,f\,x\,C_p\,(p')^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (11)$$

and

$$p' = C_p \sqrt{\frac{15\,P}{f\,x}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (12)$$

In the same way formulas may be deduced for wooden cogs by assuming the width of tooth equal to .6 p'—a common value.

TABLE NO. 2. VALUES OF COEFFICIENT $C_s = \left(\frac{t^1}{t}\right)^2$

No. of Teeth in Gear.		Value of Coefficient $C_s = \left(\frac{t_1}{t}\right)^2$			
		Involute.		Cycloidal.	
Exact.	Intervals.	15°.	22½°.	12 Tooth Base	15 Tooth Base
12	12	.77	1.00	.79	.56
13½	13— 14	.83	1.08	.85	.70
15½	15— 16	.87	1.18	.92	.83
17½	17— 18	.92	1.29	1.00	.94
20	19— 21	.98	1.42	1.08	1.04
23	22— 24	1.02	1.56	1.18	1.11
27	25— 29	1.10	1.72	1.29	1.18
33	30— 36	1.18	1.87	1.38	1.26
42	37— 48	1.29	2.03	1.48	1.35
58	49— 72	1.42	2.16	1.60	1.45
97	73—144	1.56	2.29	1.73	1.56
290	145—Rack	1.72	2.30	1.87	1.69

In these formulas, as previously stated, the effective or working height of tooth was used instead of the total height, which is always somewhat greater, but as the strength of the tooth varies with the square of its thickness, and as this value increases below the working depth circle more rapidly than the increased height of tooth due to the fillet, we are justified in neglecting that portion of the tooth between the working depth and root circles, assuming that a suitable fillet has been used at the base of the tooth.

The formulas here presented will give a close approximation to the working strength of a gear tooth when the pitch and number of teeth are given, provided we know the working stress in the material of which the tooth is composed. This is, however, such a variable factor that it is possible only to suggest limitations covering general conditions; and the judgment of the designer, based on a knowledge of the conditions under which the gear is to work, must modify the general values, as occasion may require. Indeed, it is much more satisfactory to the designer to use a general formula involving a choice of stress than to employ an entirely empirical rule in which the stress is unknown, and which offers no opportunity for variation under varying conditions.

Professor Unwin assumes that for cast iron f may have the three values 9,600, 6,100 and 4,300 pounds per square inch for "little shock," "moderate shock" and "excessive shock," respectively, for those gears in which, from inaccuracy of form or fixing, the pressure may come on a corner of the tooth. For carefully fitted gearing, however, in which the pressure is assumed to be distributed along the whole width of the tooth, he suggests 4,350 and 2,780 pounds per square inch, "the latter to apply to cases where there is some vibration and shock."

In selecting these values he assumes that the load is divided between two pairs of teeth in all cases, so that the pressure on each tooth is 2-3 P. Multiplying these several values of f by 2-3, we obtain the stress in pounds per square inch, assuming the total load to be carried on one tooth.

TABLE NO. 3. VALUES OF STRESS f FOR CAST IRON (UNWIN).

	Little Shock.	Moderate shock	Excessive shock
Inaccurate gearing	6300	4000	2850
Well formed gearing.....	2875	....	1840

It will be noted that the real stress in the well formed gearing may be considerably less than that given on account of the influence of increased thickness of tooth at the base.

Professor Reuleaux, in his Constructor, states that the dimensions of gear wheels must, for the same pressure on the teeth, be increased to meet shock in proportion to the increase in initial velocity.

For slow-running gears, however, he neglects this action and divides gears into two classes, viz., hoisting gears and transmission gears, and includes under hoisting gears all those having a linear velocity at the pitch circle of not more than 100 feet per minute, and under transmission gears all those running at a higher velocity. For hoisting gears he recommends a fiber stress of about 4,200 pounds per sq. inch for cast iron, and states that an increase of one-fourth in the permissible stress would reduce the pitch only 7 per cent., but on the other hand, he notes that too low a value of f causes unnecessary increase in the

size and weight, not only of the gears, but also of the bearings, frame work and other parts of the machine. In assuming this value of f Reuleaux does not consider any increase in strength due to increased section at the base of the tooth, although he states that "the actual stress is properly somewhat less, because the thickness of the tooth at the base is usually more than one-half the pitch as assumed in the formula."

For transmission gears Reuleaux states that the fiber stress should be taken smaller for a given force P, as the circumferential velocity increases since the dynamic action of shock and vibration also increases.

For cast iron he recommends,

$$f = \frac{9{,}600{,}000}{v + 2164}$$

in which v is the velocity of the pitch circle in feet per minute. For steel he states that f may be taken 3 1-3 times, and for wood 6-10 times the value thus obtained.

Arranged in tabular form this gives results as shown in Table IV.

TABLE NO. 4. VALUES OF SAFE STRESS f FOR TRANSMISSION GEARS (REULEAUX).

VELOCITY IN FEET PER MINUTE.	100	200	400	600	800	1000	1500	2000	2500
For cast iron f = ...	4240	4060	3744	3473	3238	3034	2620	2302	2068
" steel f = ...	14112	13020	12467	11565	10782	10103	8725	7665	6886
" wood f = ...	2544	2436	2246	2083	1943	1820	1572	1381	1240

Mr. Wilfred Lewis in his excellent paper on Gearing, read before the Engineers' Club of Philadelphia, in 1893, makes the following pertinent remarks:

"What fiber stress is allowable under different circumstances and conditions cannot be definitely settled at present, nor is it probable that any conclusions will be acceptable to engineers unless based upon carefully made experiments. In the article referred to,* certain factors are given as applicable to certain speeds, and in the absence of any later or better lights upon the subject Table V. has been constructed to embody in convenient form the values recommended.

"It cannot be doubted that slow speeds admit of higher working stresses than high speeds, but it may be questioned whether teeth running at 100 feet a minute are twice as strong as at 600 feet a minute, or four times as strong as the same teeth at 1,800 feet a minute. For teeth which are perfectly formed and spaced, it is difficult to see how there can be a greater difference in strength than the well-known difference occasioned by a live load or a dead load, or two to one in extreme cases. But, for teeth as they actually exist, a greater difference than two to one may easily be imagined from the noise sometimes produced in running, and it should be said that this table is submitted for criticism rather than for general adoption. It is one which has given good results for a number of years in machine design, and its faults, such as they may be, are believed to be in the right direction."

TABLE NO. 5. SAFE WORKING STRESS f FOR DIFFERENT SPEEDS (LEWIS).

SPEED OF TEETH IN FEET PER MINUTE.	100 or less.	200	300	600
Cast iron.....	8 000	6 000	4 800	4 000
Steel.....	20 000	15 000	12 000	10 000
SPEED OF TEETH IN FEET PER MINUTE.	900	1 200	1 800	2 400
Cast iron.....	3 000	2 400	2 000	1 700
Steel.....	7 500	6 000	5 000	4 300

If the formulas and coefficients presented by the writer are followed in the determination of the proportions of gear teeth the permissible stress per square inch should be real, not apparent. If no allowance is made for the increased thickness of tooth in gears having more than twenty-four teeth, the apparent allowable working stress may be considerably greater than the real, as determined from the thickness of tooth near the base; this accounts in a large measure for the high stresses assumed to be carried by some gears. Thus, for instance, a gear having sixty-two teeth, 3.6-inch pitch and 15-inch face runs at 280 feet per minute and carries a load of 22,240 pounds. The working stress f is assumed to be 7,250 pounds, but if the increased section on the working depth circle be taken into consideration the working stress is reduced to 5,200 pounds.

In the same way a gear, 3-inch pitch, seventy-six teeth, 13-inch face, runs at 1,045 feet per minute, and carries a load of 4,350 pounds. The working stress is assumed to be 3,700 pounds per square inch, whereas it should be only 2,375 pounds if the increased section of tooth be taken into account.

An important consideration which the designer must bear in mind in selecting a suitable value of f is the possibility of serious consequences due to failure. Where the breaking of a tooth would result disastrously, it is evident that a low value of f, or a better material, must be employed; on the other hand, cases will frequently arise in which the circumstances would indicate the use of a high stress with perfect safety.

While high rim speed does not necessarily imply shock, yet the effect of shock is more liable to be disastrous if high speed obtains, and for this reason the stress should diminish as the speed increases. Another reason for this is found in the fact that the stress is increased, due to the centrifugal force set up in the metal itself, which varies at the square of velocity; while this is not very great at ordinary speeds, it is still of sufficient importance to be considered for high speeds.

The influence of change of load is well understood in engineering construction, and usually a factor of safety is adopted for such loads from two to three times greater than would be the case for a steady or dead load, the value depending largely upon the range of stress involved.

There seems good reason to believe that a similar relation modified by the effect of speed should obtain in selecting a suitable value for the working stress in gear teeth.

From the foregoing considerations, then, it would appear that the working value of f should be chosen both with reference to the velocity and to the character of the acting force.

With this in view the writer has deduced the following formula for working stress:

$$f = \frac{50000}{k + \sqrt[3]{V}}$$

in which f = the allowable stress in pounds per square inch;
V = velocity of pitch point in feet per minute;
k = a constant having the values.
5 for little shock,
10 " moderate shock, and
15 " violent shock.

TABLE NO. 6. VALUES OF STRESS f FOR CAST IRON (FLATHER).

VELOCITY IN FEET PER MINUTE.	VALUE OF STRESS IN POUNDS PER SQUARE INCH FOR,		
	Little shock.	Moderate shock.	Violent shock.
100	5200	3400	2550
200	4600	3150	2400
400	4050	2900	2250
600	3700	2700	2150
800	3500	2600	2050
1000	3300	2500	2000
1400	3100	2350	1900
1800	2900	2250	1840
2400	2700	2150	1770
3000	2600	2050	1700

From this formula Table VI. has been calculated, and if offered as a guide to the designer in selecting suitable values of f under varying conditions.

* * *

P. T. Dodge, a patent attorney of New York and Washington, has presented to Cornell University what is said to be the costliest piece of machinery ever constructed. It is the original Paige type-setting machine, the only one of its kind ever built, and is said to represent an expenditure of $2,000,000, although it is not to be supposed that this sum went for designing and machine work. It is also said to consist of 19,000 parts, which, according to the statement made to us by a mechanic who has worked on the machine, were beautifully finished and very accurately fitted, the whole machine constituting a remarkable piece of machine work, aside from its costly features. The machine would set separate type, distribute and justify perfectly. It was a failure in a commercial sense, for even after the first machine was perfected it was impossible to build the machine so that it could be sold. The machine occupies floor space 11½ by 3½ feet, and has a maximum height of 6½ feet.

* Paper by John Cooper, in Journal F. I., July, 1879.

SHOP TALKS WITH YOUNG MECHANICS.—9.

STANDARD OF MEASURE—GAUGES.

W. H. Van Dervoort.

Modern methods of manufacturing interchangeable machine parts necessitate the extensive use of standard gauges. A standard gauge is a fixed caliper, the distance between the measuring surfaces representing a certain definite portion of the British Imperial yard, which is also the standard in the United States. This British yard is the distance between two very fine rulings on polished gold plugs inserted in a certain bronze rod commonly referred to as bronze No. 1, and carefully guarded within the walls of the Houses of Parliament.

As changes in temperature affect the length of the bar, the distance between the lines is standard at 62 degrees Fahrenheit. The rulings on the gold are so fine that they are hardly visible to the naked eye, necessitating the use of the microscopes for making comparisons.

A number of copies of the British yard were made for distribution among the different governments and to preserve as reference in case of accident to the original standard. Of these only two, bronzes No. 19 and No. 28 were exactly standard, the

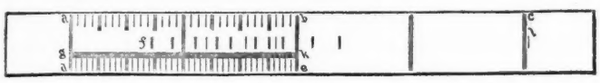


FIG. 155.

others varying slightly from the original. Both of these are retained by the British government as exact representations of their standard of measure.

By act of the British Parliament, the Imperial yard was legalized in 1855, and the following year copy No. 11 was presented to the United States Government. At 62 degrees Fahrenheit No. 11 is .000088 of an inch shorter than bronze No. 1, and is exactly standard at 62.25 degrees. The temperature at which these bronzes are standard is usually referred to rather than the amount of their error at 62 degrees Fahrenheit.

The exact subdivision of this standard yard into its thirty-six equal divisions, each of which represents one inch, and the still further divisions to the fraction of an inch was a task of no small proportions. Sir Joseph Whitworth obtained the inch subdivision by making first three end measure test pieces, each equalling as near as possible one foot. Keeping these pieces continually of equal length they were worked down until the three when placed end to end exactly equalled the standard yard. By continued subdividing the foot pieces were reduced to inches. For the fractional division of the inch he depended on the accuracy of a screw. The enormous amount of time and work required to make this subdivision can hardly be appreciated, and when completed the uncertainty of maintaining the test pieces as standards due to the wear on the measuring faces by the repeated trials necessary in the comparing of working test pieces renders the method unsatisfactory. These uncertainties were eliminated in the method proposed by Prof. Wm. A. Rogers and so ably carried out by himself and the Pratt & Whitney Co. This method substituted for the end measure test pieces a graduated steel bar, thus making a line measure reference. As working gauges are made of tempered steel, it was important that the reference bar be made of the same material, since the effect of changes in the temperature affects to an unequal degree the expansion of different metals. As repeated comparison with the bronze standard would not be practical or even desirable, it was decided by the Pratt & Whitney Co. to construct a hardened steel bar, to be graduated and compared with the official standard, and that this bar, when its precise relation to the standard was established, might thereafter be used as a standard with which to compare steel gauges and tools necessary in the production of accurate interchangeable work.

Of the five bars graduated and compared with bronze No. 11 the shortest, a tempered steel bar, 6 inches long and ½ inch square in cross section is the one about which manufacturing interests most center. The upper surface of this bar is ground and polished to a perfect plane, and 4 inches of its length graduated, as shown in Fig. 155. The subdivisions along one edge are inches, half inches, quarters, eighths and sixteenths,along the opposite edge, tenths and twentieths and through the center a series of lines representing the exact bottom diameter of U. S. standard thread gauges from ¼ to 4 inches, with a band of rulings immediately below of 2,500 per inch. These lines were cut with a diamond, and are exceedingly fine, being not greater than 1-25,000 of an inch in width.

A complete description of the method employed for ruling and comparing these bars with the standard, bronze No. 11, together with a description of the Rogers bond comparator, is given in a book published by the Pratt & Whitney Co., entitled "Standards of Length and Their Practical Application."

The most careful investigation of this steel line-measure bar shows it to be but five millionths of an inch longer at 62 degrees Fahr. than 1-9 of the Imperial yard. The greatest error in any of the subdivisions does not exceed 1-50,000 of an inch. By means of microscopes provided with micrometer eye pieces the reading or comparing of lines on these finely ruled bars can be made with the greatest exactness, and at the same time not impair in any respect their accuracy by wear, as is the case when using end measure standards.

As a measuring machine, the essential features of this remarkable instrument and its working may be briefly stated as follows: Two accurately lapped surfaces corresponding to the anvil and measuring point of the common micrometer serve as caliper points, between which the articles to be measured are placed. The measuring point is secured in the end of a sliding plunger, which carries in a suitable position a small plate, upon which is ruled a very fine reference or setting line. Back of, and capable of motion, parallel with the plunger is a carriage mounted on suitable guides. Secured, adjustably, to this carriage are two powerful microscopes, each provided with micrometer eye pieces, by means of which variations in the setting as small as 1-60,000 of an inch may be read. The standard steel bar is so placed that one of the microscopes passes over its surface when the carriage is moved along its guides. A spiral spring attached to the measuring plunger in a suitable manner provides equal pressure of contact between point and work, as between point and anvil.

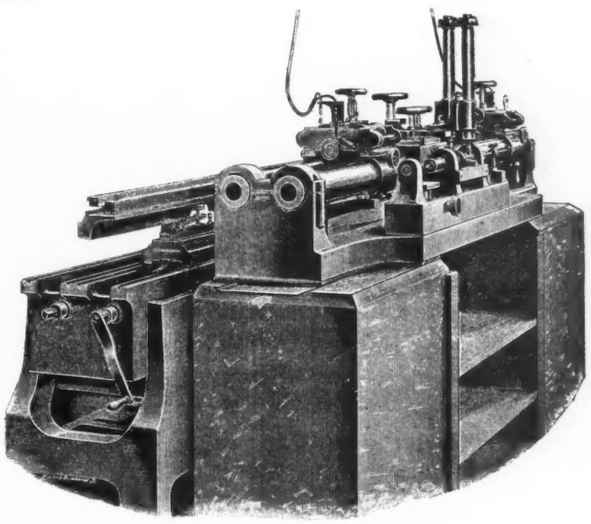


FIG. 156. ROGER-BOND COMPARATOR.

In operating, the surfaces of the measuring point and anvil are brought into contact, the carriage is moved to the point at which one microscope shows exactly over the line on the plate on plunger, and the other microscope is adjusted to read exactly on the initial line of the standard bar. The plunger is now moved back, and the article to be measured placed between the points. The carriage is now moved along its way until the first microscope again reads on the line on the plunger. The other microscope will have moved over the standard bar an amount equal to the distance between the measuring points, and the reading on the bar, with the aid of the micrometer eye piece for the minute subdivision, will give the exact measurement in terms of the standard.

Although desirable to maintain the temperature uniformly as near 62 degrees Fahr. as possible, it is not necessary in the measuring of hardened steel articles, as the coefficient of expansion for these and the standard bar is the same. It is, however, very

important that both the article being measured and the standard bar be at exactly the same temperature.

In Fig. 156 is shown a general view of the Roger-Bond comparator, owned by the Pratt & Whitney Co.

With this machine measurements guaranteed correct to the 1-50,000 of an inch are continually being made in the production of the standard gauges of this company, and dffferences in dimensions as small as 1-100,000 on an inch readily indicated. It will be understood, however, that measuring and indicating are vastly different matters, as a difference in dimensions too small to be read with any degree of accuracy can be readily de-

FIG. 157.

FIG. 158.

tected. Thus we see that at great expenditure of time and labor a working standard, the reliability of which is beyond question, was produced.

Standard gauges cannot be classed as measuring instruments, as they do not determine the amount of variation when they do not fit the work. If the work is very nearly to size, however, the tightness of the fit will give an idea as to how much the work is over or under size; this based on the judgment of the operator, of course.

In Fig. 157 is shown a plug and ring gauge. The diameter of these gauges is, as stamped on them, 1½ inches, as near as human skill can make them. The particular gauges shown are guaranteed correct to the 1-50,000 of an inch. It is perhaps difficult for you to conceive of a dimension so small. Take 1-100 of an inch, which is the smallest graduated space commonly put on the ordinary steel scales. Imagine this small distance divided into ten equal parts, and then one of these portions into fifty equal parts, and you really have something small in the way of distance. Although so nearly of the same size, by skillful manipulation the plug may be made to enter the ring. Both surfaces must be wiped perfectly clean and coated with fine oil. When entered the one will move over the other with great smoothness and surprising ease, but if the motion is stopped for

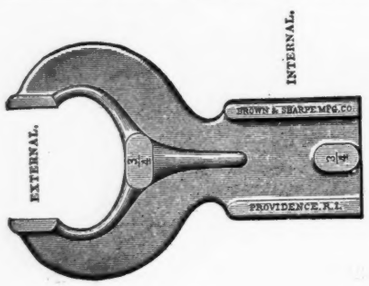


FIG. 159.

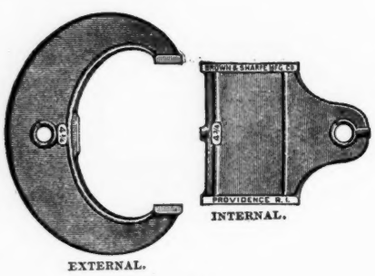


FIG. 160.

only an instant they will set, and so firmly that nothing short of a sharp blow with a mallet will start them. There can certainly be but little room for lubrication between these surfaces, and the supposition is that when motion stops the little globules of oil flatten out so thin that the surface of the metal coming into contact grip each other. The effects of temperature are clearly shown by first cooling and then warming slightly the plug. In the first case it enters the ring readily and in the latter it cannot be induced to start. These gauges are made in any desired sizes, from ¼ to 6 inches.

In Fig. 158 is shown an end measure test piece. This little piece of hardened steel measures exactly between faces the distance stamped on it. The small hole in the side is to receive a small wooden handle with which to manipulate it, as the heat from the fingers would otherwise affect its length.

The accuracy of the method of making and the final measuring of these test pieces was shown most conclusively by the severe test of the committee of the American Society of Mechanical Engineers appointed to investigate the subject of standards and gauges. The test consisted of the placing end to end in a groove planed in a heavy block of cast iron Pratt & Whitney end test pieces of miscellaneous lengths aggregating a total length of 12 inches, between suitable stops which were so adjusted that a ¼-inch end measure, used as a try piece, was held with just sufficient friction to allow it to move easily. Twelve different sets, all of which had been resting on the bock of iron sufficiently long to attain the same temperature were tried between the stops. Each set was made up of different combinations of lengths using in each case the same ¼-inch try piece. With one exception these twelve sets were found to be of uniform length, the try-piece fitting with practically the same friction in each of the eleven sets. In the one case where the try-piece was quite loose, it was found that the total length of the pieces was 1-10,000 of an inch short, most of this error being in one piece, which had been previously rejected because of a defective face.

Fig. 159 shows a form of caliper gauge, double end pattern for gauging both internal and external dimensions. For the larger sizes the forms shown in Fig. 160 are used, as they make lighter and more convenient tools. For general shop use, where constant reference is to be made to them, these gauges are preferable to the plug and ring forms, being lighter, cheaper and more convenient. With the ring gauge it must be applied to the end of the work, while the external caliper gauge can be used at any point in the length of the work.

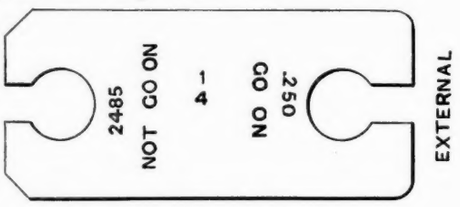


FIG. 161.

Good judgment must be exercised in the use of all accurrate gauges. They must never be forced, as in that case excessive wear results, and they soon become unreliable. A standard ring is not intended for measuring anything except work that by careful methods has been brought to the gauging diameter. It should move freely over the work without forcing, and yet close enough to show no shake. When it is considered that 1-2,000 of an inch is the difference between a tight and a loose fit, the results possible with gauges of this character become evident. In using the external caliper gauge, if forced over the work not only wear to the faces, but springing of the jaws will result, both of which destroy the accuracy of the tool. Never attempt to use this or any other gauge on rotating work, as the faces will catch and draw over work several thousandths over gauge sizes.

In the machining of nearly all work, a certain amount of variation from the exact standard size is permissible, the amount of this variation depending on the nature of the work. If, for example, the variation allowed in the marking of a large number of like parts was .001 of an inch from exact size, the economical production of these articles would not permit the loss of time in making them unduly accurate.

The gauges shown in Figs. 161 and 162 illustrate external and internal limit gauges. One end of the external gauge shown must pass over work not exceeding .250 of an inch in diameter.

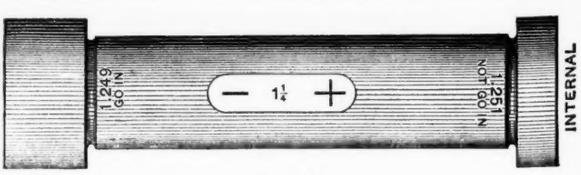


FIG. 162.

The opposite end, which is but .0015 of an inch smaller, must not go over. In this case the limit is all under the exact size, while in the internal gauge shown in Fig. 162 the limit of .002 is half below and half above the standard size.

In Fig. 163 is shown U. S. standard thread gauges, external and internal. By an extremely accurate method, the threads of these gauges are ground after hardening, which leave them high-

ly finished and correct as to pitch and angle. The standard pipe gauges are illustrated in Fig. 164. The use of standard thread gauges enables the manufacturer of pipe, fittings, screws, etc., to maintain standard threads and sizes by referring all taps and dies to the standards.

The corrective gauge shown in Fig. 165 is used for testing the correctness of caliper gauges. To insure against errors arising from the use of gauges which, through wear or accident, have become incorrect, it is important that they frequently be referred to the corrective gauge. Caliper gauges which have worn on the faces until they are oversize, can be brought up to standard by pening lightly the outer edge of the crescent.

The use of standard gauges and micrometers have done much to systematize, improve and lessen the cost of many lines of manufacture. With our present excellent systems of gauging it would be quite possible to build the many parts of a steam engine or machine tool in any many different shops throughout the country and to assemble these parts into a perfect working machine.

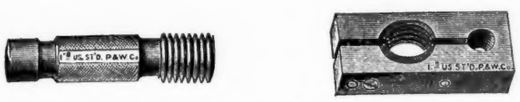


FIG. 163.

In the manufacture of gauges the cost depends very largely on the degree of accuracy obtained. A set of cast iron plugs and rings for common work can be ground correct to one-fourth of a thousandth without much trouble. But when of tempered steel and the error reduced from 1-4,000 to 1-50,000 of an inch, the cost of production due to this great refinement increases many times.

Gauges that are to be finished to the highest possible degree of accuracy are, after being hardened, ground to nearly the required

FIG. 165.

dimensions, and then stored away to "season." Gauges that are finished immediately after hardening will frequently crack open several months later. It has been found, however, that if the gauge is allowed to stand for about a year before the finishing work is done on it, it is not apt to crack thereafter. Gauges are made standard at 62 degrees Fahr.

* * *

There has been some criticism of the award of a $500,000 contract to the Brown Segmental Co., of Reading, Pa., for twenty-five 5-inch and twenty-five 6-inch Brown segmental wire guns for the United States government. It is claimed by the manufacturers, however, that these guns will stand from 50 to 100 per cent. heavier charges than the built-up gun, and that they will endure a greater number of rounds. It is stated that if Sampson's fleet had been supplied with these guns he could easily have bombarded Santiago from his position off Morro without injury either to the guns or the ships.

* * *

That the best things are not always as consistent as they might be is a source of comfort to us all, and was well illustrated by the last number of our bright little contemporary, "The Locomotive." This paper, as is well known, deals with boilers, not locomotives, but the last number contained only one article upon boilers and gave most of its space to "The International Date Line," "The Voyage of the Cruiser 'Amazonas'" and "Prophetic Cockroaches." It was an interesting number, nevertheless.

ENGINE FOUNDATIONS AND PIPING.

PROPORTIONS AND DETAILED DIRECTIONS FOR THE BEST RESULTS.

THEO. F. SCHEFFLER, JR.

Engine foundations should be just as carefully proportioned as the engines themselves to obtain the best results. The best results consist in steady regulation and stability of the whole engine, with as little pounding as possible, and they can be obtained only by having well-made, generously-proportioned foundations. The writer is well aware that there are engines on the market to-day, where great care and attention has been given to thoroughly counterbalancing the revolving and reciprocating parts, so that they will run steadily and smoothly set upon four pins, and will not jar or shake off the pins when running the ordinary number of revolutions per minute, even though not bolted down. This is all right theoretically, so far as it goes, but it will not do for large and heavy duty engines in actual practice, and especially when they are very heavily loaded.

FIG. 164.

The writer has taken as a basis for this article a 16 x 24 engine foundation, the proportions of which have been satisfactory in every particular instance where the soil and bottom of foundation has been satisfactory to build on, and where the drawings were carefully followed.

The depth of foundation for this size engine should be at least 7 feet 6 inches, unless there is a good rock bottom found before reaching this depth, so that the engine anchor bolts may be anchored directly to the rock; otherwise, if there is no such rock to anchor to, and the above depth has not been made, there will not be enough weight to the foundation to hold the engine down properly. The length of the foundation on the shaft end of the bed, measured from the center of the shaft to the end of the foundation, should be equal to the length of the foundation, measured from the cylinder end of bed to the center of the shaft. The correct proportion for this length of the foundation under consideration is about seven and one-half times the stroke of the engine. In a great many cases this length has been very much diminished on the shaft end of the bed; but there is no better place to throw in brick than on the shaft end, as here is where the weight is required, especially if there is a very heavy flywheel and shaft. On the engine under consideration, however, the above length has been cut down somewhat, since the engine has a cylinder support at the cylinder end of the bed, which gives a long foundation over all. The length at the crank-end of the foundation is about 9 feet, which is about four and a half times the stroke of the engine. Ordinarily on short stroke engines the above proportions which have been given should make the ends of the foundations equally divided on each side of the center line of the engine shaft, making the center of the shaft in the center of the apex of the foundation.

The width of the foundation must, of course, be governed by the style of foundation, or the style of engine to be used. On a 13 x 12 high speed engine, the width at bottom should be equal to eight strokes of the engine; in this case, the stroke of the engine being 12 inches, multiplying by 8 gives 96 inches for width at bottom. When an outend box is used, the width can be somewhat narrower directly under the engine, but the foundation directly under the outend box should be connected to the engine foundation proper, as otherwise one may settle more than the other and throw the engine shaft out of line. In putting in large foundations, it is a good plan to provide a manhole at the bottom to enable a man at any time to get at the foundation anchor bolts, and in case one should break a new one could be replaced very easily. The size of manhole should be about 16 inches square. In putting in a new foundation the bottom should be well pounded before the bottom of foundation is commenced, and if in any way found soft, it would be well to drive piles if the softness extends down to any extent. Another way is to obtain good cypress wood and tar the pieces and lay them horizontally with engine and then commence the bottom of the foundation on this; underneath the wood, however, it should be perfectly level, and earth or gravel well pounded in. This will

give a good, solid starting point for the foundation proper.

In commencing to build an engine foundation, a wooden templet should at first be obtained of the exact outline of engine bed, giving the location of center line of cylinder and also of shaft; on top of the templet the distance should be marked, giving the height of engine bed, from bottom of bed to center of shaft; the exact location of anchor bolts should also be given as well as diameter. Having obtained the templet, it should be located exactly level in its proper plane, and the anchor bolts of the engine should be inserted through the templet, having first put the anchor-bolt washers on the bolts. Having located the templet correctly as to height above the floor line, the building of the foundation can be commenced. Generally the first foot in depth of an engine foundation is composed of small broken stones, which are well cemented together by cement. This mixture is called concrete, and after the cement has hardened, it becomes like rock. This makes a splendid foundation for the brick, which should not be soft, but a good quality of hard burned brick, yet not so hard as to crumble or break easily. The brick should be laid in a good quality of hydraulic cement, and if a cap-stone is to be used, it should be not less than 8 inches thick, or it will be liable to crack through the center of the bolt holes. It is a good plan to use a 2-inch pipe around the anchor bolts, near the floor level for a distance of about 2 feet below the floor, so that the anchor bolts will enter the holes in the bed easily, and then when the engine has been made perfectly level at all points, build a dam all around the edge of the bed on the foundation cap-stone with either fire-clay or putty, and pour sulphur or lead under the bed, so as to fill up the space between the cap-stone and the bottom of the bed. In leveling up the bed thin wedges should be used between the bed and cap-stone, thus allowing a space for running in the lead. This space should not be over 1-16 or 3-32 of an inch for the best results. The engine bed can be leveled up by placing a spirit level on top of the lower side of the bed, and also by trying it on the main journal. Having leveled up the engine and filled space with lead, screw up the anchor-bolt nuts somewhat, and again try level to see that nothing has moved out of line. While we are mentioning the main engine foundation, the details of outend box foundation will not be amiss, as well as a brief description of the box itself.

The general design of outend box should be about the same as the main bearing of the engine, with the omission of the gibs for adjustment. The gibs are generally omitted in this bearing, as it is independent and can be moved back and forth at pleasure in order to have perfect alignment with the shaft. The bearing, however, should have a sole-plate underneath it, with a wedge between the sole-plate and bearing, so that adjustment can be obtained by raising and lowering the bearing. This is not always done, but it is advisable on very large and heavy duty engines, especially where the end of shaft is to connect with a gear or clutch for transmitting power; also if the foundation of outend box bearing should settle a little more than the engine foundation proper, the wedge can be brought into service and raise the bearing to its normal level at the same height it was before the foundation had settled, thus obtaining perfect alignment of the shaft. On this size engine the anchor bolts for the outend box should be 1¼ inches in diameter; the hole in sole-plate for the anchor-bolt is generally cored out ¼ inch larger than the bolt at top of boss, and about ⅜ to ½ inch larger at bottom, after the sole-plate is in position on the foundation, and in perfect alignment with shaft. The above holes should then have lead filling poured between the anchor bolts and inside surface of the holes. This method makes a good, solid job. The taper of the wedge should be from ½ to ¾ of an inch per foot for the best results, and the length of the wedge should be about 6 inches longer than the bottom length of journal box. This is done to obtain about ¼ to ⅜ of an inch adjustment vertically. Slots should be cored in the wedge at each end, equal in length to the diameter of the bolts holding the bearing to the sole-plate and the horizontal motion that wedge has. In this instance the diameter of bolts is 1¼ inches and 6 inches for horizontal motion of wedge, given 7¼ for length of slots. For neatness a small oil reservoir can be placed directly under the side bosses on the journal, to catch the dripping of waste oil. This idea will prevent that sordid and really injurious constant flow of oil down the side of the foundation side walls.

If a cap-stone is used between the bottom of the sole-plate and the foundation, it should be of good, hard quality, like granite. It has been the writer's experience that if stone like soft, white sand-stone is used, it absorbs the oil, after which it is liable to crack if the anchor bolts are tightened up unequally.

On very large and heavy-duty compound engines some manufacturers furnish large, cast iron sub-bases directly under the outend box sole-plate. When this is done it makes a solid and substantial job. The cap should have a lip at each end to lock the bearing to it, the same as on main journals, which will prevent any horizontal springing. A small tongue should be planed on the bottom of the wedge, and extend downward into sole-plate, to keep the wedge in perfect alignment with center of the journal; a tongue should also be placed on the bottom of the bearing. These tongues can be 1 inch and ½ inch deep. In commencing to build the foundation for the outend box, the same plan followed as for the main foundation. On high speed engines the extra length on the crank-end previously mentioned is absolutely necessary for quiet and steady running of the engine. This extra length will in all cases provide an ample foundation base and mass in line with the motion of the engine piston.

The foundations for the above engines cannot be too well constructed. The very best brick and cement should in all cases be provided, and great care taken to be sure that the masonry is well bonded, closely laid and properly leveled up. The concrete should in all cases be composed of an approved brand of Portland cement, clean, sharp sand and clear, broken granite or trap rock of a size that will pass through a 2½-inch ring, and it should be proportioned and mixed as follows: One part cement, two of sand and five of stone, mixed each time by careful measurement.

Engine Piping.

While we are considering the engine foundation, it will not be out of place to mention something in regard to connecting engine to the boiler. Too much cannot be written about the proper design or construction of boiler and engine plants either for electric lighting station or for power plants, or, in fact, any kind of station; they both go together.

Considerable space and convenience may be saved and obtained by leaving the design to the manufacturers, as well as economy secured in pipe connection between boiler and engine.

For small plants, where there is but one boiler and one engine, it is customary to set the engine about 12 feet to 14 feet from the boiler, and put an 8 or 12-inch division wall between the boiler and engine room, so as to keep all dust accumulating from ashes out of the engine room. For the best results engines should be set longitudinally with the boiler, so that direct pipe connection can be made. This is also, in the writer's opinion, the best method on account of the expansion of the steam pipe. In plants where there are three or four engines to be connected to a battery of boilers, and the engines are placed either directly in the rear, or in front of the boiler fire-room, there will be considerable waste of steam pipe, which will amount to quite an item on account of the steam pipe being large in diameter to accommodate the volume of steam from the boilers to the engines.

The steam and exhaust pipes must be as large as the inlet and outlet holes, respectively, on the steam and exhaust chest. Have the steam pipe as short and direct from the boiler to the engine as possible. If it is desirable to use the exhaust steam for heating or other purpose, put the connection in the exhaust pipe by a three-way valve, so that, if necessary, the steam can be exhausted direct, and around this three-way valve make a by-pass, in which place a back-pressure valve, so that in case the back pressure should exceed four or five pounds, the regulator valve will open and relieve it. Usually a feed water heater is used, and probably the best and most convenient place to locate a feed water heater is on top of the boiler side walls, placed at right angles to the boiler, and set horizontally, the heater being supported at each end by a cast iron leg or bracket anchored securely to the side walls by anchor bolts. This method of locating the feed water heater has been used on several large boiler settings which the writer has designed. Any person who is in any way familiar with boiler settings knows how hot it is over the top of boilers, and can readily see the advantage of locating the feed water heater above them. From 5 to 10 degress of

additional heat will thus be obtained for the feed water over that supplied to the heater when the heater is set up in the ordinary manner. To set the heater on top of the boiler will cost a little more, but it will soon pay for itself by being more economical. The exhaust pipe leading from the engine to the main exhaust pipe that leads to the heater should be constructed so that there will be a minimum amount of back pressure. To connect the exhaust pipe to the main exhaust in a satisfactory manner, a lateral branch "Y" connection should be made where the pipe meets the main exhaust pipe. A valve should be located on the pipe leading to the main exhaust pipe, so that in case any one of the engines is stopped for repairs where there are several engines connected to the same exhaust pipe, the exhaust steam from the other engines will not back up into the steam chest in case the valve of the engine is disconnected from the engine. This valve will be appreciated when the engine is undergoing repairs. The "Y" connection also does away with the short and sharp angle connection, and makes a freer passage for the exhaust steam. Before connecting the steam pipe to the engine permanently, blow steam through it, in order to clear it of any burrs or thread ends which may be left in the pipe. A good separator should also be connected to the steam pipe near the engine; also a sight feed lubricator. It is a very good plan to provide a shut-off valve on the steam pipe at the boiler, so that in case of any accident at the engine throttle the steam can be shut off and thus prevent further accident or loss of steam, or probably a shut-down of the plant. On a number of cheap plants this valve is not always provided.

* * *

A PECULIAR BOILER ACCIDENT.

Editor MACHINERY: Enclosed you will find "photo" of a singular explosion of the mud-drum to a Babcock & Wilcox boiler, at the works of the Ithaca Drop Forge Co., at Ithaca, N. Y.

The explosion occurred at 9:15 a. m., Monday, Sept. 5 (Labor Day). The engineer and a steam fitter had been repairing on of the pipes from the boiler to the "nickel room," and after attaching the pipe to the boiler, the fire was "hustled," or crowded, to get up steam to start the engine, so the polishers and hammermen could go to work.

FIG. 1. BEFORE THE EXPLOSION.

A number of the men had just left the boiler room and gone to their several stations, when there was a sharp report, and in another instant the drum exploded, blowing down the side wall and scalding the engineer so that he died the same day at 8 p. m.

The boiler room was a one-story building, 16 x 50. The instant the explosion occurred the boiler room door blew shut, causing some delay in the escape of the engineer from the room. The external scalds of the engineer were not considered dangerous, but it is thought that the delay in his getting out caused him to inhale the steam, thus producing death.

The boiler had been thoroughly cleaned, and the tubes from the mud-drum to the boiler were new about two weeks prior to the accident. The work was done by an old, experienced boilermaker, and everything pronounced in good condition. No cracks were noticed in the drum, and, as you will see, the fracture is entirely apart from all connection, the hand-hole, blow-off pipe and connecting tubes being all intact, as though nothing had happened.

The writer is an old engineer, and has seen a great many queer things happen to boilers, but this is the queerest of them all.

One theory, by a prominent engineer, is that the rapid firing caused the water to be driven from the drum, and then the water

FIG. 2. SHOWING THE BURSTED MUD DRUM.

hammer, which would occur when the water came back, caused the first report heard by the fireman, and this hammer started a crack; then the rapid cooling caused uneven expansion, enlarging the crack and then the explosion.

The drum was 12 inches outside diameter and 26 inches long, of cast iron, from 5/8 inch to 3/4 inch thick; the cracks all showed new, and the iron is apparently as good as it was the day it was cast.

In the "photo" of the boiler front you will see that the boiler and coal bin lean together; this is caused by the ground or foundation settling; the boiler had been for some years in this way. It was about 5 inches out of plumb. The boiler was about fifteen years old (40 HP.).

Ithaca, N. Y. H. C. FAIRBANKS.

[It should be stated that the cast iron mud drum was used only on the earlier types of the Babcock & Wilcox boiler, and that wrought steel is now employed for all the parts.—Editor.]

* * *

The American consul-general at Shanghai, China, reports that some machinery made in the United States for a mint at Chengtu, China, has been set up and tested. It was delivered to the Chinese authorities at the head of navigation on the Yangtze River, and its subsequent treatment was characteristic. The machinery was in boxes, as shipped from the factory. It was left on the bank of the river for a month, so close to the water that when a slight flood came the boxes were under water, and the muddy water deposited sediment until the boxes were full of mud. In this condition they were taken to Chengtu, being about two months on the way. When the boxes were opened they were found to be caked up solidly with mud, except where the rats had dug holes and made nests while the boxes lay at the arsenal. The machinery finally was gotten into condition to be put together, but the dies were so badly rusted that they would be condemned in an American mint. The Chinese, however, made no objections to the imperfect places in the coins, apparently thinking them additional safeguards againsts counterfeiting. Some part of the machinery were entirely ruined and others were so badly rusted that they cannot last long.

A WORD ABOUT APPRENTICES*.

Speaking of apprentices in general, I think it will be conceded without much argument that the more intelligent a class of mechanics becomes, the higher the standard of the work which they produce, and the higher the moral tone and self-respect of such class. It seems to me necessary for the foundrymen and manufacturers generally throughout the country to encourage and employ apprentices, for the sake of supplying the places of journeymen who fall out of the ranks through sickness, death, etc., and to supply the necessary molders for new industries. It, therefore, seems desirable to me that young men of education be induced to seek this vocation as a means of livelihood.

If this is the fact, the question then arises as to how we may secure such apprentices. The experience in our works has been that it is very difficult to induce young men who have passed the grammar schools, or schools of equal or higher standard, to embark in the molder's or other trades. There seems to be no difficulty in getting plenty of applicants of an uneducated class, but our experience in the past has been that this class are not willing to submit to the necessary discipline, and if they are, they are usually dull and slow to learn. On the other hand, we have found that where we could get applicants who had been through the grammar schools, or other equally good schools, they appreciated the fact that discipline must be enforced, were always willing to live up to the rules, were anxious to advance themselves, and were students in their new vocation, as well as they must have been at school, in order to have passed through it. Having acquired the habit of study, we find that they apply the same thought to their work, can readily grasp the instructions given by the foreman, and are not satisfied with merely getting through with their work as a stint. On the other hand, they study the conditions and take great pride in the character of work which they are turning out. From this experience, we have reached the conclusion that we shall try to confine ourselves to a class of apprentices who have at least reached as high a standing as to pass the sixth grade in the public grammar schools, and, if possible, who have graduated from the grammar school or some school of equal standing with the public schools of our city.

In casting about for the reason why we do not have applicants from this class of young men for apprenticeship we reach the conclusion that the inducements offered are too small, and therefore decided to offer a better scale of wages and inducements, for several reasons. First, in many instances the parents of would-be applicants of this kind are mechanics who have had a great struggle to give their children the benefit of a grammar school education, and who, having other children coming along who are gradually absorbing more of their means, feel that as soon as the boy has passed the public school, he should be in condition to be self-supporting, and cannot therefore afford to contribute much if any toward the support of the boy for a further term of two or three years while he is passing through the earlier stages of his apprenticeship. On the other hand, it is natural, and in one sense creditable to the boy, that he should wish to be self-supporting, and also only natural that he should desire to earn enough so that he may have some small amount of spending money. These desires, we think, often blind the boy, and perhaps the parent as well, to the fact of the reward when the apprenticeship is done, and the possibility that the boy may find some employment which will earn him something like $1 or even $1.50 a day at once, although it may be nearly as high an amount of wages as he will ever reach, decides him to go in this direction.

I know a number of young men who attended public school with me when a boy who are to-day in positions such as street car drivers, conductors, teamsters, etc., who I remember as bright boys in their studies. Some of them have told me that the mistake of their life was in not having learned a trade, and I feel sure some at least would have been able to have earned much more money than they are now earning, and perhaps been high in the offices of some good institution, had they learned a trade thoroughly.

I am sometimes inclined to agree with a friend who once asserted that no one ever learns anything from experience. Be that as it may, young people are slow to profit by the experience of others, and since the fact remains that we do not seem to get many applications from desirable apprentices, we set the foregoing reasons down as being responsible for the fact, and adopted the following standard:

We refuse to take any applicant who is not to appearances physically sound, cannot bring good recommendations as to character, has not passed through the public schools at least as far as the sixth grade, or of some equivalent educational institution, and is willing to serve three months on trial before signing articles for a four-years' apprenticeship; the three months of trial, however, being counted as so much time on the first year.

A year is to consist of 300 days of 10 hour each, and for
The first year we pay 7½ cents per hour, or $4.50 per week.
The second year we pay 10 cents per hour, or $6 per week.
The third year we pay 12½ cents per hour, or $7.50 per week.
The fourth year we pay 15 cents an hour, or $9 per week.

The week consists of 60 hours, and it is our policy to give the apprentices full weeks, even if the machinery is not running, by placing them at such times on work that can be done without the use of power.

This scale of wages was made with the idea that it would be nearly if not quite the equivalent that any would-be applicant would be offered if he were going out to find work of a general character, and would therefore be attractive to applicants. We offer a further inducement in the fact that we agree to give each apprentice at the end of his apprenticeship the sum of $100 as a bonus for the faithful performance of the contract entered into between us. We are of an opinion that from an economic standpoint we might do better with journeymen at journeymen's wages, but we also believe that it is a duty which we owe to ourselves, and to the manufacturing fraternity to keep up an apprenticeship system which will develop good journeymen, and we feel that if we do this, and other manufacturers do it, we shall be able to supply yourselves in the future with mechanics of such class as will be better qualified for the work demanded of them, and in this manner be indirectly repaid for the outlay.

We require of our apprentices that they attend night drawing schools, so that they may become competent to make ordinary sketches, and to help them to learn how to read drawings, and it is our policy to interest ourselves in these matters to a large extent in order to encourage the boys in the greatest degree. I am pleased to say that we always find the class of apprentices of which I have been speaking not only willing, but anxious to avail themselves of such facilities, and that the foremen take great interest in such cases, and are always willing to help the good work along.

Not only has our experience with apprentices who have passed the grade of public schools been satisfactory, but in instances where we have had apprentices who are graduates of the technical or training schools, we have found their progress rapid and their ability equally greater, and for this reason we place such graduates in a separate class, offering them apprenticeship positions which omit the first year's work, and starts them on the second year. We feel that we can do this for two reasons: namely, they have, through practice received at the training schools, acquired some ability, which lessens the amount of attention they need from the foreman; second, a superior educational training and discipline incident thereto fit them to progress faster and accomplish the same results, or better, in the shorter time.

* * *

At the recent meeting of the American Association for the Advancement of Science, Prof. Thomas Gray, of Terre Haute, Ind., referred to a peculiarity in the elastic properties of a specimen of flat bar steel, which was exhibited at the meeting. The bar was 0.41 inch wide, by 0.063 inch thick. It had been heated in the smithy fire, and allowed to cool slowly in the air. When the bar was held like a flat spring, and bent first in one direction and then in the reverse direction, the elastic limit was found to be fully three times as high for bending in one direction as for bending in the other. The values for the bending moment at the yield point were found to be 8.9 and 30.8 inch-pounds, respectively. It seemed that the heating in contact with the hot coal in the smith's fire had produced properties on one side of the bar approximating to those possessed by cast iron—namely, greater elastic strength against compression than against extension.

* Abstract of paper read before the American Foundrymen's Association by

WHAT MECHANICS THINK.

A DEPARTMENT FOR THE EXPRESSION OF PRACTICAL IDEAS OF INTEREST, TECHNICAL OR OTHERWISE.

Write on one side of the paper only, and when sketches are necessary, send them. No matter how rough the sketches may be, we will see that they are properly reproduced.

HE WAS NEVER STUCK.

In the eighties I worked for a man who owned a little shop in a country town, and nearly all the work was done by the job. The reason for this was that the mill men in the surrounding country had learned that "Old Ebb," as they called him, had a faculty for making work expensive unless he was penned down to a price before he commenced a job. But Old Ebb also had another faculty for getting around a "snag," as we used to call it, and he would never let a job go away because his machines had not capacity enough. When business got dull the old man would quietly slip out of sight for a day or two, and when he

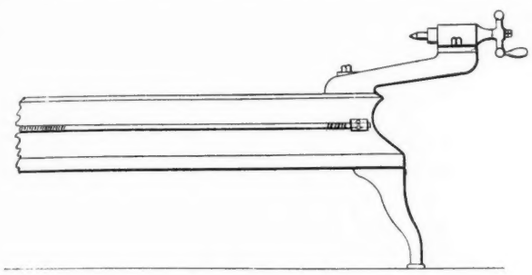


FIG. 1.

returned he would nearly always have business enough to keep us hustling for some time. It was on one of these trips that the old man got a sticker for a job. He took an order for about 200 feet of shafting, the lengths to be 14 feet exactly. The largest lathe we had would take 12 feet 7 inches between centers. I got the job to center and turn the shafting and fit the couplings. It being midwinter and the iron being behind the shop under 3 feet of snow, I did not fancy the job much and I immediately reported to the old man that the lathe wasn't long enough into 17 inches or so. He set me to work on something else for the time being, but I felt in my bones that sooner or later I would have to dig that iron out of the snow and turn the shafting, although I did not just see how I could do it. We ran by steam power, and took turns firing up in the morning. There were six of us, including the old man, so it came out just even. It was my turn to steam up next morning and build the fires in the two big stoves in the shop. It was about five when I arrived at the shop, and I found the old man busy on a pattern. I got the fires going and stood around, waiting for the old man to give me some excuse to ask what he was working on. I soon had the information without the asking by the old man starting in in this way: "You shovel out that pile of iron after you get back from breakfast, and get it around into the shop. I will have that lathe ready by day after to-morrow to turn it."

FIG. 2.

Sketch No. 1 shows how he made the lathe long enough. Sketch No. 2 shows a special center he made to drive the shaft with. Cutting a ½-inch slot across both ends of each shaft and commencing at the head stock, we turned up the shafting and faced the coupling by taking a light cut. The remark the old man made when I suggested that he was going to make more expense than he would get for his job, was this: "No job ever goes out of this shop because we can't do it, and you fellows who want to keep your job just keep this in mind."

E. St. Johnsbury, Vt. B. F. GROUT.

TWO SIMPLE RULES.

I send the following rules, which may be interesting, especially to apprentices. First, a short rule for compound gearing:

A set of compound wheels are wanted for a lathe having a 4-pitch lead screw to cut a screw having five threads per inch. Assume any two convenient wheels—in this case we will use 40 on the spindle, and, say, 80 on the screw—then multiply the first (40) by 5 (the pitch of the screw to be cut), and arrange the work as shown. Multiply the other gear selected by the pitch of the lead screw (4), and then divide the two products by some common divisor, 5 in this case: 40 on spindle; $40 \times 5 = 200$; $200 \div 5 = 40$, gearing with wheel stud; $80 \times 4 = 320$; $320 \div 5 = 64$, second stud wheel; 80 on screw.

Now, if we wanted to cut a ⅝ pitch screw and our lead screw had 4 threads per inch we would proceed as follows: Place a 64 on spindle; $⅝ \times 64 = 40$; $40 \div 2 = 20$; $4 \times 32 = 128$; $128 \div 2 = 64$. Place a 32 on screw.

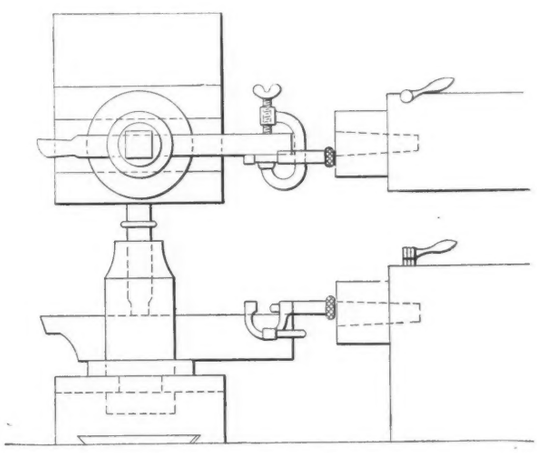
Second, here is an easy rule for pulley speeds.

Speed of driver × diameter of driving pulley ÷ by the speed of driven = diameter of driven pulley. Thus, if the line shaft runs 200 revolutions per minute with a 12-inch pulley, and the machine to be driven must run 450 turns, what diameter pulley will be required for the machine? Following the rule, we have $200 \times 12 \div 450 = 5$ 1-3 diameter of driven pulley.

Or, suppose the line shaft runs 250 turns with a 12-inch pulley, machine to run 125 turns. As this is a ratio of 2 to 1, a 24-inch pulley will be required. G. L. R.

USING A MICROMETER FOR A DEPTH GAUGE.

I am sending a sketch of what I consider a very handy use of the micrometer. I have a great number of irregular forms to recess in work held in the lathe chuck, which has to be very accurate; and not being able to use any of the depth gauges now on the market, for the reason that you cannot get them into a very narrow slot or determine the correct depth of a very irregular form, I had to devise some other means for measuring the depths, and adopted the method shown in sketch, which I have found to be the most satisfactory depth gauge I have ever used on any work within its limit. The principle reason for its popularity with me is that, no matter what sort of a form or recess you are making, you never have to withdraw the tool until the operation is finished. In all other gauges I am familiar with, you



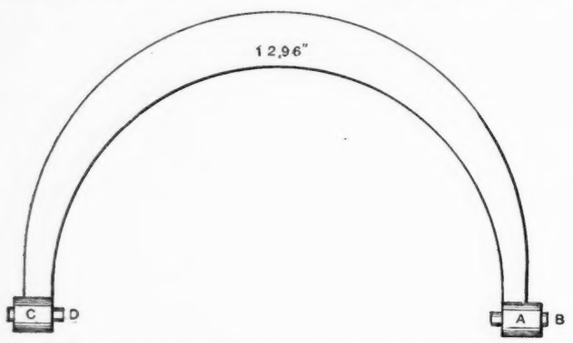
have to withdraw the tool quite frequently to try the gauge and see if you have reached the proper depth. I fasten the micrometer on the tool I intend to do the recessing with, then take a light cut across the face of the work, set the micrometer back against the tail-stock and start the cut, at the same time revolving the barrels of the micrometer back so as to keep in contact with the tail-stock until I have entered the desired depth. There is no cut and try about it. It will be noticed by the sketch that if the tool had been set at an angle the micrometer would not rest against the tail spindle, but against the face of the tail-stock. This is immaterial so long as the point of contact does not move. I also use the same principle very successfully on the shaper, when I clamp the micrometer on the head instead of the tool and allow the barrel to follow the slide. J. R. S.

A FORM OF CALIPER GAUGE.

The sketch shows a caliper gauge which has the feature of adjustability when worn, and which is convenient and inexpensive to make. The one shown is about the shape and proportion of some which were used in turning a lot of thirteen-inch shells for Uncle Sam's big guns.

As will be seen, the gauge is marked 12.96 inches, which was the size to which the contractor had to make the shells, which were afterwards hardened and finished at the governmen arsenal. This would indicate that the body of the projectile is at least .04 inch under the nominal size.

The gauge is a forging with the bosses A and C. These bosses are drilled and reamed out for the hardened plugs B and D, which are forced in with as heavy pressure as practicable to



use, and which are then lapped to the size desired. When worn it is an easy matter to press one or the other of these plugs in and again lap to the standard size.

The same principle can also be used on some forms of inside gauges.

Corning, N. Y. FRED E. ROGERS.

ANOTHER USE FOR SOAP.

The following may be of interest to those brought up in fear of anything like soap in the hardening water.

Our tramp had some small tools which required to be very hard, and at the same time very tough. After heating in the usual way, and forcing through a bar of common laundry soap, the tools were in a condition to cut glass, and at the same time tough enough to stand severe usage.

The writer has used this method on small drills, scrapers, etc., with excellent results. No drawing is required. J. T.

TRUING WORK IN THE CHUCKS.

The following method can be used for truing up work at once without the use of chalk or the hammer. Bend a piece of sheet metal about 5-16 inch wide by 7 inches long, as shown in Fig. 1. Clamp the work lightly in the chuck, clamp a tool in the tool

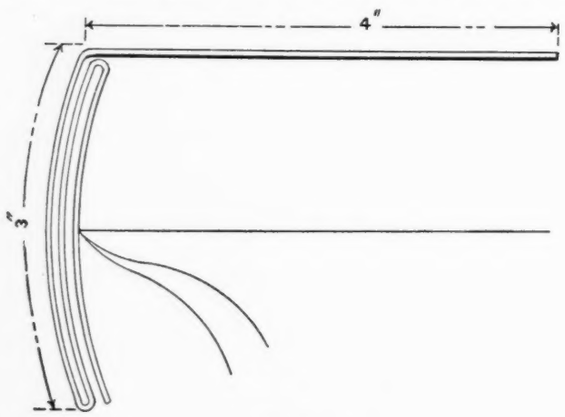


FIG. 1.

post, and start up the lathe. Bring the tool up against the piece in the chuck, and at the same time hold the bent metal strip between the tool and the work, so as to prevent marring or digging into the work. Put a drop of oil back of the strip, and if the work has been finished before chucking, a piece of copper wire flattened on the end and bent into the right shape will serve the purpose of the strip better than a piece of sheet metal; otherwise sheet tin will do.

Having trued the work by bearing against it with the tool while the lathe is running, tighten the jaws of the chuck and the work will run very nearly true. Where the work is to be drilled after truing up, a centering tool can be placed in the

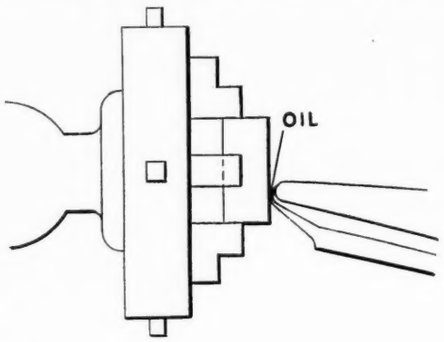


FIG. 2.

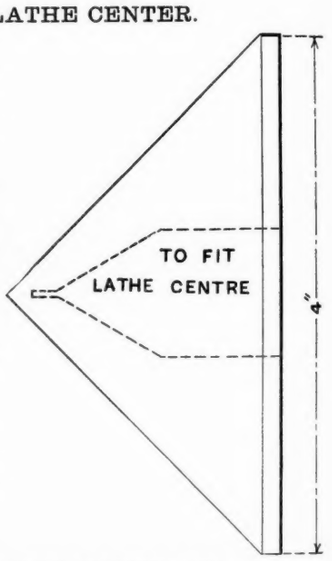
tool post, and after using it for truing the work, it will be all ready for countersinking to start the drill.

This is a small thing, but I consider it a great time saver, as in truing up work, even with a test indicator, the work must be struck with a hammer just right to get it true, which is seldom done by striking it once or twice. Of course, where extreme accuracy is wanted, it would be well to test it after the operation. ARTHUR MUNCH.

Brainard, Minn.

LARGE LATHE CENTER.

I enclose a sketch of a large center to be used on the lathe centers for large work. It is a useful device, especially in jobbing shops where pipe threads are usually cut in the lathe; also, when turning the outside of long bushings, etc. The angle and size is left to the judgment of the user. I consider that with a loose center the friction is less than on the large, stationary centers, and it can easily be made, finished in the chuck at one setting, and can be hardened if thought necessary.



H. R.

Trenton, N. J.

TENSIONS NOT CONSTANT.

"Retsel's" letter in October MACHINERY appears to demand that I shall produce authority for saying that the sum of the tension is not constant. Now, I was well aware that there were loads of authority to support all "Retsel" has written on the subject, but I know of no proof that it was true. I had seen horizontal belts spring the shaft more when transmitting their maximum power than when running empty, and I judged this to be sufficient ground for criticising "The Principles of Belting," I have just completed a search for further testimony, and have found what I hope may induce authors to change their text books in conformity therewith.

On page 881, of Kent's Pocket Book, the following statement is to be found:

"The discovery of Messrs. Lewis and Bancroft (Trans. A. S. M. E., VII., 549) that the sum of the tensions on both sides of the belt does not remain constant, upsets all previous theoretical belting formulae."

Mr. Wilfred Lewis presented an exhaustive report to the society in 1885 of experiments made, which he sums up as follows: (I quote only what refers to tensions.)

That the sum of the tension is not constant, but increases with the load to the maximum extent of about 33 per cent. with vertical belts.

That with horizontal belts the sum of the tensions may increase indefinitely, as far as the braking strength of the belt.

In regard to the arc of contact, I can only say that I have found a belt to slip on the driver—when a slip occurred—when the pulleys were of the same material and in like condition. Any one can satisfy himself in this respect by putting the cone belt on the fastest speed of lathe, drill or miller, then lock the cone and put on the driving belt, and note where the slip occurs. One does not have to get the aid of the calculus to determine the value of the arc of contact, when the large pulley is the driver, except in extreme cases, unless I fail to discover the true inwardness of the action of belting.

When theoretical formulae do not consider practical conditions, we cannot expect trustworthy results therefrom.

Youngstown, O. J. H. DUNBAR.

(We submitted Mr. Dunbar's letter to "Retsel," giving him an opportunity to reply in this issue, and thus shorten the debate.—Editor.)

THE REPLY.

Replying to Mr. Dunbar's letter, the tests made by Messrs. Bancroft & Lewis seem to leave no room for argument as to whether the tensions remain constant. The tests were made under working conditions, and show plainly that they do not, and I am only too glad to have some satisfactory evidence in the matter, even if it hits me. Mr. Dunbar will note, however, that my bone of contention was not as to whether the tensions are constant, but as to whether his argument that while the tensions are constant for vertical belts they are not constant for horizontal belts, was sound.

It remains to inquire whether the tensions vary enough with a change in load to make the common theory untenable. There is very little in machine design that can be held down to any theory or hard and fast rules. The theories do not always "hold water," and there are so many conditions to be taken account of that it is not strange that they do not. It is well known that the common theory of the strength of beams is not correct, but a better one has not been devised, and so the common theory is used, with such modifications as the experienced designer thinks best to introduce. And so with belting it is best to adhere to the old theory, with suitable modifications to suit the conditions, unless it can be proved to be positively misleading, or until a better one is originated. Referring to the Bancroft and Lewis tests, it will be seen that where the sum of the tensions increases largely there is an excessive slipping of the belt, such as should not occur in proper running. In fact, if one considers the life of the belt as of value, he should so proportion the belt that it will not slip excessively. Good running calls for a slip, at ordinary speeds, not exceeding 5 to 15 feet per minute, which will produce a coefficient of friction of leather on cast iron of perhaps three tenths.

In Table I. of the tests referred to are two tests made on cast iron pulleys under normal conditions that appear to indicate that so far as the tension part of the theory is concerned, it is satisfactory enough under ordinary working conditions. These tests were made at only 800 feet per minute belt speed, and the slip at this speed should be low. Taking this into account, the tests show that the sum of the tensions does not increase more than 5 to 10 per cent. at the outside under fair working conditions, which is as close as we can hope to come with any belting rule.

Regarding the effect of the arc of contact, if I was in error, the error was on the safe side, and I still believe that the rules given are safe ones to follow. "RETSEL."

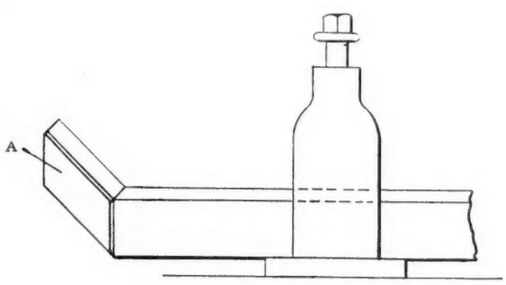


FIG. 1.

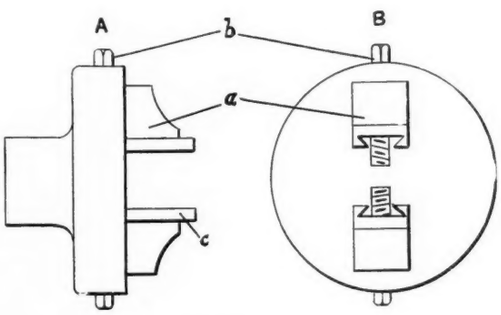


FIG. 2.

A FEW NOTES.

Fig. 1 is a polishing tool, consisting of a piece of flat iron that will fit into the tool-post of a lathe, bent to an angle of about 45 degrees, and the end covered with leather (a). The carriage can be run up close to the work, and a piece of emery cloth inserted between the work and tool, which enables the workman to get a good polish; it is valuable on anything that has square corners, etc.

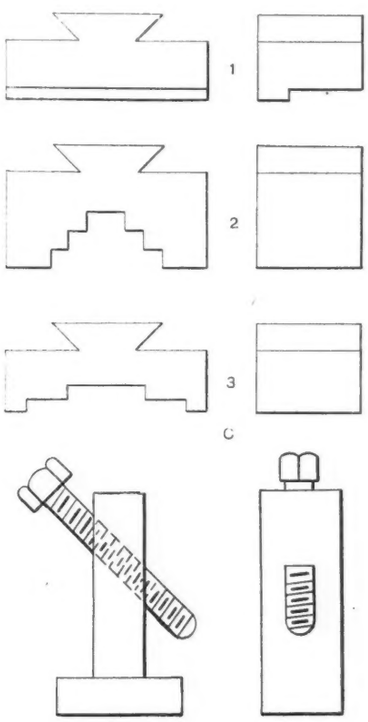


FIG. 3.

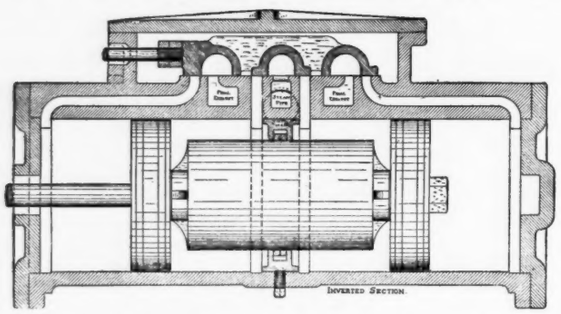
Fig. 2 represents an ordinary two jawed chuck, with one screw, one end with right and the other end left thread (b). The ends of the jaws (a) are dovetailed, to permit the removable or false jaws C 1 and 2 and 3 to be inserted or removed. The jaws C are merely a tight fit in the jaws (a) and are held by no screw or other means. Work can be held in a chuck of this kind that is often done on the shaper, as small brass heads, etc. We have it in use on a turret lathe in the shop where I work. The false jaw may be grooved or formed in any manner so as to hold special work.

Fig. 3 is a dog or clamp for a planer. The bottom is fitted in the T-slot of the planer bed, and the set screw is tightened down on the work. The set screw runs through at an angle of 45 degrees or less, according to the inclination of the user.

WOOLF TRUNK ENGINE.

The novel compound pump, described on page 61 of October MACHINERY is in some respects similar to the Woolf trunk compound engine.

The Woolf engine was built some years ago, but, if my information is reliable, is not now in use, owing to the excessive leakage of steam past the large central packing ring. The great



area that the ring has is not only difficult to keep tight, but the friction arising from the packing, when it is tight, is excessive.

Though I doubt it, the concern building the engine described in your columns may have overcome those difficulties.

Auburn, N. Y. J. B. HALL.

* * *

PRACTICAL PROBLEMS.—1.

A CHANCE TO GIVE YOUR BRAIN A LITTLE EXERCISE.

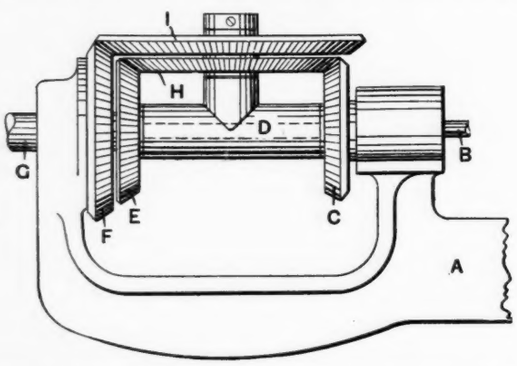
We have received a number of replies to Mr. Rogers' chuck problem too late to be used in October, and as three solutions were given in that number, we will not take the room in this issue for the later solutions. The active interest taken in this and various other problems that have been published from time to time led us to ask Mr. Rogers to send us problems every month, solutions to which will be published the following month, thus making these problems a regular feature of the paper that will not only be an interesting one, but of real benefit to subscribers who have time to give the problems a little thought and study. For this issue Mr. Rogers has sent two problems, which are published herewith, and we will ask that solutions be sent to us as soon after the receipt of the paper as possible. There is no assurance that a solution received late in the month can be published.

1.—A Gear Problem.

The sketch shows a mechanical movement, which is known as a differential epicyclic train.

It is a combination of five bevel gears mounted as shown in the frame A, which forms a support for the two shafts G and B. These shafts have a common axis, but are independent of each other.

The driver of the train is the right-hand gear C, which is mounted on shaft B. This shaft has an extension to the left, as



shown by the dotted lines, which forms a bearing for the sleeve D. This sleeve is free to revolve on its bearings, and carries a stud, on which are mounted the two gears H and I. These are fastened together and turn together as one gear, but E and F are separated. F is stationary, being fastened to the frame A, and E is mounted on the shaft G, thus forming the driven gear.

Now, when C is turned it transmits motion to H and I, but as F is fixed, the sleeve D with the gears is forced to swing about its axis, and thereby transmits a certain amount of movement to E.

Gear C has 31 teeth.
Gear E has 32 teeth.
Gear F has 40 teeth.
Gear H has 50 teeth.
Gear I has 60 teeth.

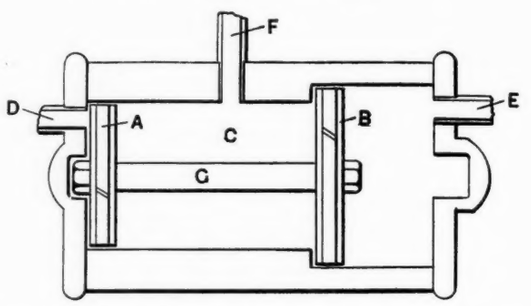
Now, will some of the readers of MACHINERY who are skilled in mental gymnastics follow the evolutions of these gears from start to finish, and tell us what the ratio of the driver is to the driven, and also if the direction of rotation of the shafts G and B is the same or opposed?

2.—A Question of Pressure.

The differential piston is a device widely used in one form or another. The air pump to be seen on almost every locomotive contains one which forms part of the reversing device.

Reducing valves and similar appliances also make an extended use of the principle.

The word differential explains its action, which is simply that the effective pressure to be derived from is the difference in the pressure exerted on the two outer or inner sides of the pistons A and B. These pistons being connected by the stem G move as one piece. It is clear that with equal pressures per unit of area the piston will be driven in the direction of the greater force by an amount equal to the difference in areas of A and B multiplied by the pressure per unit of area. It will also be seen that with a certain pressure on B, a higher pressure will be required on A to force the piston to the right.



Now, to cause a little thought and study on this question the following problem is proposed, to which answers are solicited:

The two pistons in the sketch A and B are of 4 and 5 inches diameter, respectively, and they are subject to such pressures through the pipes D and E that the direction of the greater force is to the left, giving the position shown.

These pipes are connected to the same source of supply, but a reducing valve in pipe E reduces the pressure per square inch of area to 8-11 of that in D.

A pressure of 40 pounds per square inch being admitted to the space C through the pipe F moves the pistons to the right with a force of 141,372 pounds.

What are the respective gauge pressures per square inch of area against A and B?

* * *

The Parson's Steam Turbine Co. is now located in new works at Wallsend-on-Tyne, England, and it is stated that advance orders are such as to indicate a good business for this firm. The performance of the little "Turbinia" was so remarkable in its way, that many engineers predict a decided change in steam propelling machinery for both large and small vessels during the next few years. However that may be, the performance of some of the larger boats that this company are now building will be watched with much interest. The "Turbinia" was only 100 feet long and weighed only 45½ tons, yet she easily attained a speed of 35 knots. One of the new boats that is now being built will be twice as long as the "Turbinia," and have engines five times as powerful, or of 10,000 HP., and a still greater speed is predicted for it.

* * *

It is said that if abrasive wheels be made of a mixture of emery and carborundum they will have the advantages of the solid carborundum wheel, and also will be less brittle and can be graded more accurately.

DOINGS OF THE SHELL-FISH CLUB—III.

A TRUE FABLE. BY THE PORPOISE.

The Crab sat in the easy chair in his machine shop on the beach, wiping his glasses with a piece of sea weed, and his eyes wore a worried look. Although ages of prosperity had blessed him with pearls, the time had come when he must be relieved of the cares of supposing he knew how to manage his shop.

His desire was to secure a nice, puttied mummy to take the position of superintendent of his mechanical sand shed.

The Oyster, who happened in, suggested the Lobster, who was said to be quite old in experience, with a way for reaching results that was very desirable. The Crab's only objection, that the Lobster had claws, being removed by the Oyster's suggestion that they might be cut off, the Lobster was engaged.

Now, the Lobster happened not to be of the broiled-on-the-half-shell sort, but was a bustling, bug-eyed, green-backed fellow, with ideas of his own, and scrambled into the engine room of the place as soon as he arrived. Here he met the Turtle, with barnacles of discontent all over his back, owing to his lot in life, sweating and shoveling coal while furnishing power to the plant. "What's the matter with you?" asked the Lobster. "Nothing, but blisters on the boiler," said the Turtle. "From what?" queried the Lobster. "Engine overworked," replied the Turtle. Investigation led the Lobster to believe that the Turtle was right, and that the Clam next door, who had a chowder mill and rented power from the Crab was getting the most of it, to the detriment of the Crab's business. So the Lobster borrowed an indicator with the intention of taking continuous cards all day long from the engine to find what the Clam was getting. To do all this right the Lobster also wanted a reducing wheel. Now, the Crab had for a buyer a Sand-Fiddler, who was noted for having hysterical horrors whenever called upon to purchase anything costing over a dollar, and he found reducing wheels would cost $15, and the Lobster was called upon to explain the cause of this needless luxury. He was told that the Clam must not be disturbed, as he was paying $40 per month for 30 HP. transmitted through a 6-inch belt from that engine, and could not get more, and this $40 per month was clear profit and like finding money; upon which the Clam winked his other eye and went out and

"WHAT'S THE MATTER WITH YOU?" SAID THE LOBSTER.

bought a beer. Still the Lobster was dissatisfied. The Turtle's barnacles grew larger, the blisters in the boiler sagged deeper and the power grew weaker. One day the Lobster asked the Crab on what particular day of the month he should give the Clam the 30 HP., as at present he was giving it to him several times a day, while, did he know more of the details, he could deliver it at some particular time and have the balance of the month for the benefit of the Crab. "Explain yourself," said the Crab. "Well," answered the Lobster, "if you only give him 30 HP. per month, I must arrange the transmission differently, but if you are giving him 30 HP. per hour, per day, per month, the Clam has the big end of the cuttle bone, and you have not been 'finding' as much broth in the Clam's shell as you dreamed of." The Crab wrote the Lobster's statement down and went into executive session with the Sand-Fiddler and Oyster, at the close of which the Clam was notified to buy a wind-mill and make his own soup, as they had been in the "devil" for years, while he has been living on 300 per cent. gravy. Although the Clam was a close-mouthed fellow, he told the Lobster that if he had kept his goggled eyes in deep sea water there would be fewer

"EXPLAIN YOURSELF," SAID THE CRAB.

breakers on the beach. But the Lobster, you will bear in mind, will never be considered an angel until his back is red and his claws are gone. Then the Lobster is a good thing.

The moral of this story is never to bother with Lobsters, for, while they may save money, they create dyspepsia. Be a Clam.

* * *

LEVERS.—2.

THE CALCULATIONS OF SIMPLE LEVERS.

In the last issue it was explained what is meant by a force, the moment of a force and the principle of moments. The familiar example of the wrench was taken to illustrate these principles, so the reader would have a simple and common illustration to refer to in case of any difficulty in applying them to other examples.

It will now be shown how to solve examples of the lever by the aid of the principle of moments, and we will first consider that the weight of the lever may be neglected and that there are only two forces acting—the push or pull, which is applied to the lever, and the resistance overcome, these being balanced, of course, by the pressure at the fulcrum, which, in reality, is another force, but which need not be considered for the present, at least.

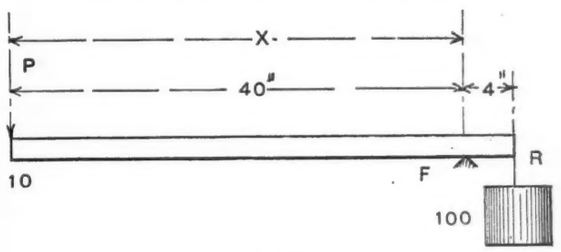


FIG. 8.

In Fig. 8 is shown a lever supported on the fulcrum F. At one end a push, P, of 10 pounds, is exerted, and at the other end is a resistance, R, in the shape of a 100-pound weight. The distance from F to P is 40 inches, and from F to R 4 inches.

The principle of moments states that when a lever is in balance the moment of the force tending to turn it in one direction must equal the moment of the force tending to turn it in the opposite direction. In Fig. 8 the moment of force P about fulcrum F, tending to depress the left-hand end of the lever, is $10 \times 40 = 400$ inch-pounds, and the moment of force R is $100 \times 4 = 400$ inch-pounds also, so that the lever is in balance.

Now, suppose that we had P, R and the distance from F to R given in Fig. 8, and that we wanted to find the distance from F to P, which we will call x. By the principle of moments we have,

Moment of $P = 10 \times x$.

Moment of $R = 100 \times 4 = 400$.

But these moments are equal; hence, $10 \times x = 400$, and what we have to do is to find the value of x. It is clear that if ten

times the distance x = 400, the distance x must be 1-10 of 400, and all we have to do is to divide 400 by 10, giving 40 inches as the distance x.

Again, suppose it were desired to find the resistance R, the other quantities being known. For convenience we will take the moment of R first, because this contains an undetermined value. This is always a good rule to follow.

Moment of R = 4 × R. (It make no difference whether the 4 or the R is written first, but it is better to write the figure first.)

Moment of P = 10 x 4 = 400.

Then 4 × R = 400 and R = $\frac{400}{4}$ = 100 pounds.

These simple examples contain all that need be known to solve lever problems where there are only two forces acting; but to make the subject still clearer a more general example will be taken.

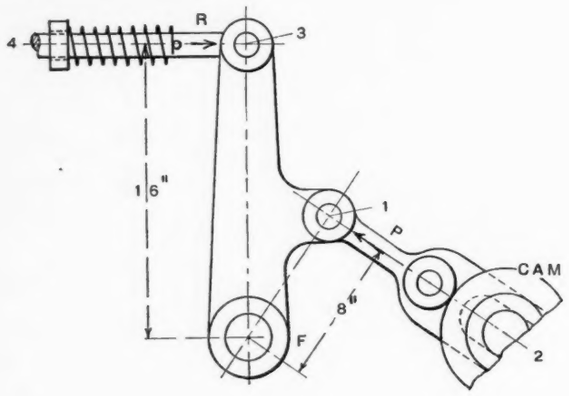


FIG. 9.

In Fig. 9 the lever shown is pivoted at F, which serves as the fulcrum. A push P is exerted by the rod at the right, which receives its motion from the cam and roller, as indicated. This push acts to overcome a resistance R, which acts along the rod seen at the left, and which may be supposed to consist of the resistance of the spring coiled around the rod and of any piece of mechanism that this rod may have to operate. Let it be required to find how great a push, P, is necessary to overcome a resistance, R, of 250 pounds. The first thing is to find the length of the true lever arms, since without these the moments cannot be determined. To do this, first draw lines through the points on the lever at which the forces act, and in the direction in which they act. Thus, the force P acts at the point 1 and the line 1—2, therefore, indicates the position and direction of this force. Likewise the force R acts at point 3 and line 3—4 indicates the position and direction of force R.

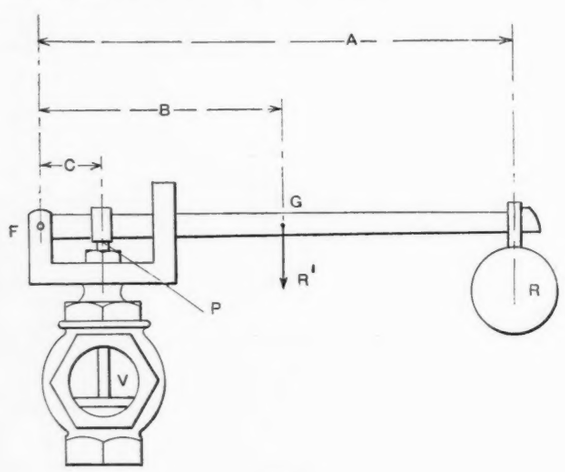


FIG. 10.

Now, the lever arm of force P is the perpendicular distance from F to line 1—2, and the lever arm of force R is the perpendicular distance from F to line 3—4. Assume that these distances measure 8 and 16 inches respectively. Then,

Moment of P = 8 × P,

Moment of R = 250 × 16 = 4,000.

8 × P = 4000

P = $\frac{4000}{8}$ = 500 pounds.

Example.—Suppose P = 400, R = 150, and the short arm = 6 inches. What is the length of the long arm? Answer. 16 inches.

The safety valve in Fig. 10 is an example of a lever in which there are three forces to be considered if we take into account the weight of the lever, as it is quite essential to do. The valve at V is acted upon by the pressure of the steam, tending to raise it. This pressure constitutes the push P upon the lever, which is resisted by the suspended weight R, and the weight of the lever, which we will call R¹. The weight of the lever is effective at the point G, the center of gravity of the lever. This point can be found by balancing the lever on a knife edge, the center of gravity being directly over the knife edge. The fulcrum of the lever is at F and the lever arms for R, R¹ and P are marked A, B and C, respectively.

Example 1.—Assume that A = 30 inches, B = 14 inches, C = 3 inches, R = 20 pounds and R¹ = 8 pounds. Find what pressure of steam the valve will carry.

Moment of P = 3 × P,

Moment of R = 20 × 30 = 600,

Moment of R¹ = 8 × 14 = 112.

For the valve to balance, the moment of P must be equal to the sum of the moments of R and R¹; for the moment of P tends to raise the lever and the other moments tend to hold it down. Adding the moments of R and R¹, therefore, we have 600 + 112 = 712, and this must balance the moment of P, or 3 × P. Hence, 3 × P = 712, and P = $\frac{712}{3}$ = 237 1-3 pounds. This last part of the operation is like the work of the previous examples.

The 237 1-3 pounds is the total pressure upon the valve and to obtain the pressure per square inch that can be carried, we have simply to divide 237½ by the area of the valve. To be theoretically exact, the weight of the valve and stem should be added to this figure (237½).

Example 2.—Suppose it were desired to carry a total pressure upon the valve of 300 pounds. With the other dimensions remaining as before, how heavy a weight R would have to be provided? Again, taking moments, taking the "R" moment first for convenience, because this is the unknown one,

Moment of R = 30 × R,

Moment of R¹ = 8 × 14 = 112,

Moment of P = 300 × 3 = 900.

The sum of the first two moments must equal the last one, but we cannot add them as they stand, because we do not yet know what the first one is. Hence we will indicate the addition as follows:

$$30 \times R + 112 = 900.$$

Those who have had a little practice with formulas will have no trouble in finding the value of R; but for the benefit of those who have not, it can be said that we subtract the 112 from 900 and proceed as in the other examples. Thus 900 — 112 = 788 and 30 × R = 788, whence R = $\frac{788}{30}$ = $26\frac{4}{15}$ pounds.

The following explanation will make the reason for subtracting 112 from 900 clear. We have found that the moment of R is 788; of R¹, 112 and of P, 900. Now, if 788 added to 112 equals 900, 900 must be 112 greater than 788 and 788 must equal 900 with 112 subtracted from it. Again, taking the formula as we have it, if 30 × R plus 112 equals 900, 30 × R must equal 900 with 112 subtracted from it.

* * *

As a contribution to the information upon the relative effectiveness of paints in preserving wood, Mr. Onward Bates calls attention in the Journal of the Western Society of Engineers to the results that were observed on an old sign board. It was erected at Harper's Ferry in 1872, and was taken down in 1896. During that time it had not been repainted and it stood where it was exposed to east winds. The original paint was probably white lead, and the letters were painted with lampblack, the shading being of some lighter color, probably a mixture of white lead and lampblack. When taken down the letters stood out in relief on the board, the wood being worn away where only protected by the white paint. Mr. Bates adds that he has always been a believer in lampblack paint as the purest of carbon paint and considers this sign an object lesson in support of his belief.

MORE "MUCKETS."

J. B. HALL.

The fixture for planing curves described by Mr. E. H. Yeakel in the September MACHINERY is only applicable to planers with two heads, and as the majority of planers have but one head, a somewhat similar arrangement that I devised and have made use of for some years is better adapted to the purpose. The outfit consists of two bolts (A and C, Fig. 1), with locknuts and nuts, one screw clamp (D) and a bar (B), as shown. These pieces

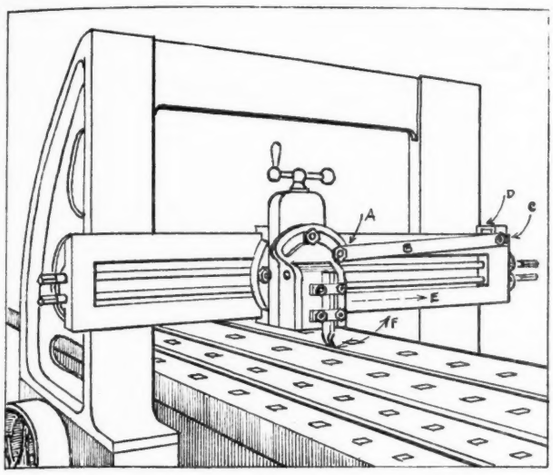


FIG. 1.

can be found around any planer. The method of operation can be plainly seen by reference to sketch. The radius of the curves (F) that can be operated by the swivel alone is limited by the length that the tool projects from the tool post; if, however, the vertical automatic feed is fed downward in conjunction with the swiveling motion, the radius of the curve can be greatly lengthened; and if the feed is upward, the curve will be much smaller in radius, than if the swiveling motion occurred alone.

Chilled cast iron makes a satisfactory substitute for mushet or other tool steel, particularly as it can be cast in a chill end on, and when cool cut to suitable length for insertion in an Armstrong tool holder. It is especially useful in machining hard iron cast in molds that were wet too much, which has a tendency to chill the surface. In grinding chilled cast iron tools, should they get red-hot so as to lose their hardness, they can be hardened by heating in a forge to a bright red, and suddenly quenching in cold water, when they will be about as hard as they were originally. Whether the quality and kind of iron they are cast from have any bearing in the matter, I cannot say, although tools made of two different mixtures (both close grained) have worked satisfactorily.

What may be in more general use than I am aware of is a wax fillet maker (Fig. 2), which is simple and very convenient, but

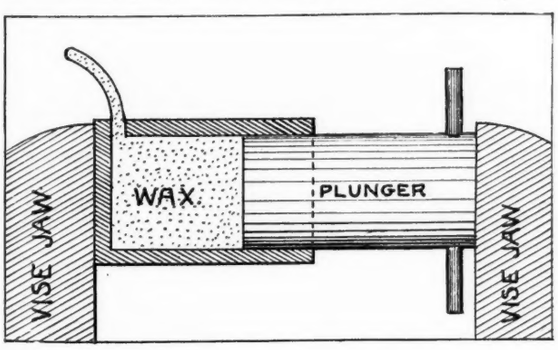


FIG. 2.

probably old. The cylinder is of any convenient size, closed at one end, with a hole the diameter of the fillet desired at the side. The plunger is a good working fit, and is fitted with preferably a T-handle for withdrawing it. Beeswax lumps are put in the cylinder, tight or loose, the piston started into it, and both put in a vise and the vise screwed up. Without any heating or other treatment, except pressure, the beeswax will flow in a solid "thread" of continuous length (unless broken off) until the pressure ceases or the cylinder is emptied.

In turning wooden patterns, instead of screwing or gluing the work direct to the lathe face plate, small pieces of paper are placed between, and the glue applied, so that it does not ooze from the paper, and stick the work and face plate together. While the heaviest cuts can be taken without fear of tearing the work loose, yet, when the work is to be removed, the insertion of a thin chisel blade in the small gap between face plate and work causes the paper to readily split, leaving part on the work and part on the face plate, from which it can be readily removed by moistening with water.

Endless belts, while not in general use on lathes or similar machinery, are of great advantage and deserve to be used more than they are. A slight drawback is the lack of means of taking up the slack, such as is obtained with dynamos and motors, instead of requiring the joint to be opened and reglued, which is somewhat of a nuisance. A means of avoiding this, while permitting the belt to be driven to its full capacity, is to glue the joint with a piece of paper the full size of scarf intervening. While the pull of the belt lengthwise is easily resisted by the paper, it can readily be split by opening a corner of the joint sufficient to get a grip with the finger and thumb, a moderate pull from which will open the joint, when the belt can be shortened, rescarfed and glued up with paper in between. Another way to open such joints is to place a moderately small sized wire in the middle of the joint and allow sufficient length to project out, so that they can be folded over the outside of belt and twisted. When the joint is to be opened, the ends of the wire can be pulled steadily by pliers, and half of the joint is soon opened, then the other half can be opened by pulling by hand or by a second wire inserted in the joint.

A burner for crude kerosene or other oils, to be used in smiths' heating furnaces, under boilers, etc., is shown in Fig. 3,

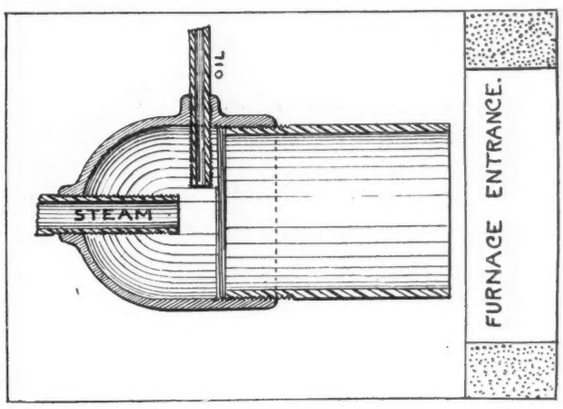


FIG. 3.

operated by steam or compressed air. It is very simple and cheap. The oil flows down the vertical pipe by gravity or otherwise, and as it leaves the pipe, the jet of steam or air from the small horizontal pipe atomizes and carries it along. The furnace is completely bricked in, except a hole about four times the area of the large pipe, into which the atomized oil enters, carrying along with it air. Inside the furnace the flame impinges on a baffle plate of fireclay, which soon becomes incandescent, assisting in the combustion. The flame is of dazzling whiteness and high temperature, while the combustion is perfect. Whether this form of "burner" is covered by a patent I am not aware, but if it is, many people are infringing, as it is a good thing.

* * *

The sixth general meeting of the Society of Naval Architects and Marine Engineers will be held at the rooms of the American Society of Mechanical Engineers, 12 West 31st street, New York, on Thursday and Friday, Nov. 10 and 11, beginning at 10 o'clock. The papers to be presented are of unusual interest, owing to the bearing that some of them will have on the recent naval actions of fleets.

HOW AND WHY.

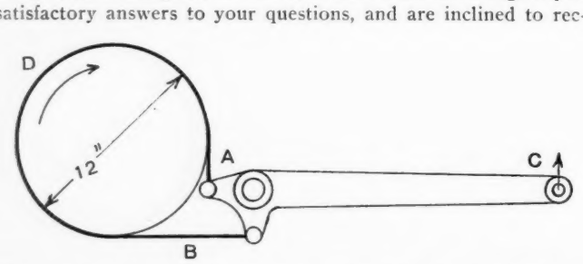
A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

33. M. R. sends two sketches of worm wheels and writes that one worm is 5-16 inch diameter and the other ½ inch diameter. Both worms run at 2,000 revolutions. How much more power will be required for the larger worm? 2. How wide should the face of a worm wheel be for a worm ½ inch diameter? 3. Would a soft steel worm and bronze worm wheel work well? A. You do not send enough information. The large worm need not necessarily take any more power if the wheel and the worm are of the right proportions. 2. Three-fourths the diameter of the worm. We would suggest that you refer to the data sheet issued with the September issue, which contains all the information that you will need about worm gearing. 3. No. Harden and polish the worm.

34. R. E. W. writes: I have had the use of some lathe patterns and have made a 14-inch engine lathe. I have taken pains with the work, and should like to know how good a lathe I have. What tests would you suggest, and what limits for accuracy would you set for a lathe of this size? A. Assuming that you want your lathe to be as accurate as you will ever require for fine work, we would suggest the following limits: The head stock spindle should line with the ways within .0005 inch in 12 inches. If there is any error it should be with the front of spindle standing toward the back, so that the lathe will bore small at the bottom of the hole rather than large. The tail-stock spindle need not line so accurately, .001 per foot or even a little more will do. A good way to test the head-stock spindle is by boring a piece clamped on the face plate. The tail-stock can be tested by fitting an arbor in the spindle and testing it with a micrometer fastened to the carriage; or you can use a test indicator if you have one. The hole in head-stock spindle should run true within .002 inch at the end of a bar 12 inches long. See if the two centers line up by turning bars of different lengths, so as to bring the tail center at different points along the bed. The diameters of the bars should not vary more than .002 inch at the ends, which will indicate that the centers are out of line with each other .001 inch. These limits will indicate what we consider to represent good workmanship, and you can adopt other tests for the lathe with these as a guide.

35. A. E. S. writes: A brake shown in the sketch consists of a steel band attached to the lever, as indicated, with a thin strip of wood between the band and the pulley. When the outer end of the lever is pulled in the direction C, the brake goes into action and goes out when released. What puzzles me is this: Is the ratio of the tensions A and B the same as in a belt, or are the tensions equal? 2. How long would it take to stop the wheel? It goes at 1,800 revolutions per minute, and the mass at the rim is 200 pounds. A. We shall be unable to give you satisfactory answers to your questions, and are inclined to recommend you to revert to experiment rather than to depend upon calculations. In any case, we should wish to have experiments performed rather than to depend on the calculations. With the lever proportioned as shown, the stress in the band at A and B would be the same when the pulley was stationary; but as soon as the pulley starts the stress at B will increase and at A decrease. We should expect the sum of the tensions to remain nearly constant. If you care to work it out, you can calculate what the tensions would be by the belting formula given in any mechanics, assuming a coefficient of, say, .1 to .2. If the wood is not well lubricated it will char. Having obtained the tensions, their difference will be the effective pull. 2. We prefer to reverse your second question, and find the effective pull necessary to stop the wheel in a stated number of revolutions. Suppose it were desired to stop it in 1,000 revolutions. Then we must equate the work done against the brake band in 1,000 revolutions to the change in the kinetic energy during that time.



Let E = effective pull.
Let R = radius of pulley in feet.
Let D = diameter of pulley in feet.
Let N = number of revolutions per second at the start.
Let n = number of revolutions before stopping.
Let W = Weight of wheel rim.
Work on brake = 2π Rn E.

$$\text{Kinetic energy to be absorbed} = \frac{2\,\pi^2\,N^2\,W\,R^2}{32.2}.$$

Equating these two values and reducing, we shall get

$$E = \frac{.049\,N^2\,W\,D}{n}$$

Substituting,

$$E = \frac{.049 \times 30^2 \times 200 \times 1}{1000} = 8.8 \text{ pounds.}$$

You would have to realize eight or ten pounds effective pull. The total pull is liable to be ten or fifteen times this amount.

36. J. T. asks: Will you kindly explain how I may calculate the horse-power of a gas engine? 2. Also, when driving through gearing and power is transmitted from a small gear to a large one, how much more work can the large gear do than the small one? A. Multiply together the area of the piston in inches, the length of the stroke in feet, the mean pressure of the gas in the cylinder and the number of explosions per minute. Divide the result by 33,000. For example, in an engine with a cylinder 10 inches in diameter by 18 inches stroke and 20 explosions per minute, we should have, area of piston = $10^2 \times .7854 = 78.54$ square inches. Length of stroke in feet = 1½. The mean pressure would have to be obtained by applying the indicator; suppose this to be 45 pounds. Then, applying our rule, $78.54 \times 1.5 \times 45 \times 20 = 1{,}060{,}290$. This, divided by 33,000 = 32.13 HP. 2. No more work. The shaft on which the large gear was fastened would turn with more force than the shaft having the small gear, but it will also turn slower. If there were no loss by friction, the force exerted by the first shaft, multiplied by its speed would equal the force exerted by the second shaft multiplied by its speed, so that the work done in both cases would be the same.

37. L. F. writes: Please tell me what the shade lines of a drawing are for? 2. What is the best rule for shade lines? 3. Should the lines be drawn outside or inside of the edge that is shaded? A. Shade lines are supposed to be drawn on the edges separating light from dark surfaces (surfaces that are in the shadow) and thus indicate on any view whether a part is raised or depressed. For example, a circle representing a hole would be shaded on one side, and where it represented a pin, it would be shaded on the other side. 2. Hold the drawing vertically in front of you, and assume the light to fall upon it from over your left shoulder. Then separate light from dark surfaces by shade lines. This resolves itself in practice to putting shade lines on the lower and right hand edges of all views. 3. It is better to draw them on the outside.

38. V. M. W. says: We are planning to make a 9-inch crank shaft and crank, and would rather force them together cold than to shrink on the crank, so as to be sure of a good and true job. All the jacks that we can muster foot up to 50 tons. Do you think they will put the crank and shaft together, and how much force would you allow? A. It is doubtful whether you can force more than .005 inch or .006 inch with 50 tons. This is a small allowance for so large a shaft, and we should rather shrink on the crank and true up the crank-pin hole afterwards.

39. M. B. writes: I am told that it takes nearly or quite as much power to reverse the driving pulleys of a planer as the

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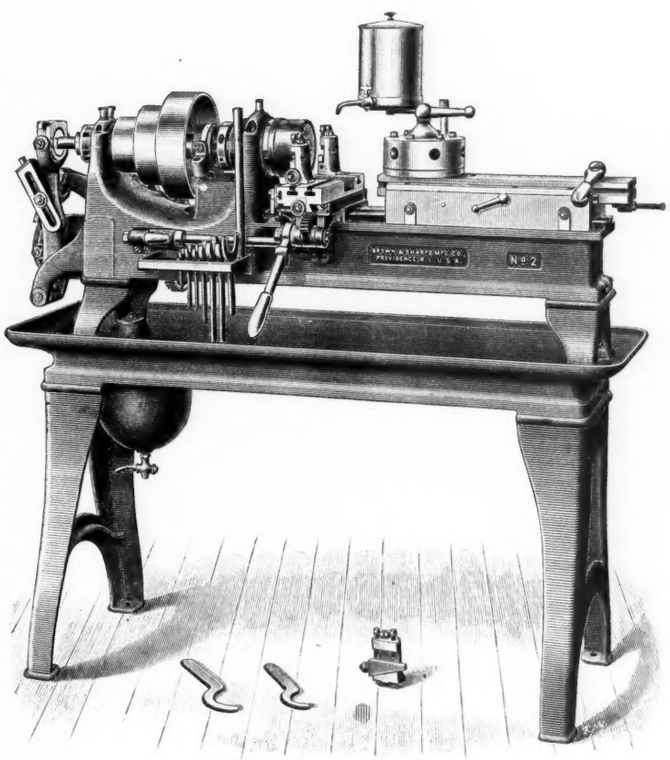

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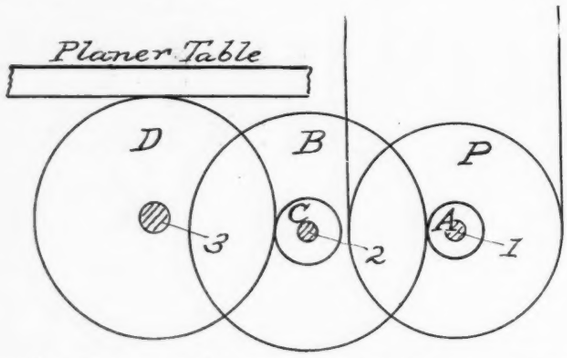
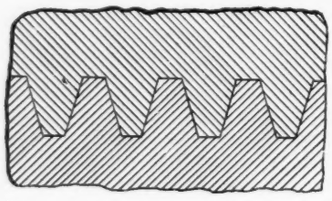
table on which there is a heavy casting. Please tell me if this is so. A. Suppose a planer to be geared so that the belt travels 50 feet to 1 foot travel of the table. The power required to reverse the pulley or the table is proportional to the square of their velocity. One pound on the rim of the pulley would therefore have the same effects as $50^2 = 2{,}500$ pounds on the table. There is good ground for what has been told you. 2. In the accompanying drawing is shown the arrangement of some planer



gearing. Please explain how to calculate the force acting on the pitch line of each gear. A. You do not show the size of the gears, and we will assume values. Suppose there is to be an effective belt pull of 100 pounds, and that pulley P is 30 inches in diameter and pinion A on shaft 1 is 4 inches in diameter. Then the stress at the pitch line of A would be $30/4 \times 100 = 750$ pounds. This would also be the stress at the pitch line of gear B. For gear C, which is on shaft 2 with B, we would have supposing B to be 28 inches and C 5 inches, $28/5 \times 750 = 4{,}200$ pounds. This is likewise the stress in D and the force with which the table is pushed against the tool.

40. F. J. M. writes: I have to use large screws, the usage being severe. What is the best form of thread for such screws, I mean for both strength and durability, as well as efficiency? A. Some sort of compromise thread will probably be the best. With the square thread there will be less loss of power than with the V thread, because in the latter part of the power applied is resolved into lateral force to split the nut. On the other hand, if it comes to a question of stripping the thread the square thread is the weakest, as may be readily seen. We should be inclined to give the preference to a form of compromise thread shown as possessing to some extent the advantages and disadvantages of both the V and the square thread. The flat at top and bottom of the thread may be greater or less than shown, and the depth may be greater or less; the proportions appear about right.



41. L. F. writes: Why does a tube bent to a segment of a circle tend to straighten when pressure is applied internally? A. A round tube bent as you say will not become straighter when internal pressure is applied. When it is elliptical in section it will straighten more or less when bent in the direction of least diameter, because internal pressure tends towards making the section round and the effect is to straighten it. If the tube were bent in the direction of its greatest diameter, it would tend to curve it still more, but in the opposite direction.

* * *

FRESH FROM THE PRESS.

THE THETA PHI DIAGRAM. PRACTICALLY APPLIED TO STEAM, GAS AND OIL ENGINES. By Henry A. Golding, A. M. I. M. E. Published by the Technical Publishing Co., Lim., Manchester, Eng., and for sale by D. Van Nostrand, Murray street, New York. 127 12mo pages. Price, $1.50.

This is a book upon entropy, and the entropy-temperature diagram, which is at the present time receiving considerable attention from students in thermodynamics. To those who have the time and disposition to specialize in this branch of science, we can recommend this book as the most exhaustive and complete treatment upon the subject of entropy that we have seen. The author attempts to make many practical applications of the entropy-temperature diagram to steam, gas and oil engines, and while they comprise interesting studies that should be taken up by any one who wishes to be well informed regarding the latest theories, we are inclined to doubt the practical utility of the subject under discussion. This is not saying, however, that the book is not as good a one as its subject will allow. We believe it is.

HAND BOOK OF CORLISS STEAM ENGINES. By F. W. Shillitto, Jr. Published by the American Publishing Co., Bridgeport, Conn. 197 pages, illustrated. 12mo size.

This book is by a practical man, and is divided into two parts, one of which treats of erecting Corliss engines and the other of adjusting Corliss valves. While few engineers have engine erecting to do, and generally have their engines placed in position for them, it cannot be doubted that every engineer would be the gainer by learning so much of this branch of the business as is contained in this little book. And especially will this chapter be valuable to young mechanics in the shop who are anxious to get outside. The chapter on valve adjustment will be valuable to mechanic and engineer alike. Directions are given for squaring the valves, adjusting governor, eccentric, dashpot and reach rods, for centering the engine, and there is attention given to the indicator diagram and its relation to valve setting.

The book closes with a few tables and memoranda, and with descriptions of several of the leading types of Corliss engines.

CATECHISM OF THE STEAM PLANT. By F. F. Hemenway. Press of "Locomotive Engineering," 256 Broadway, New York. 128 pages, pocket size. Price, 50 cents.

The bare announcement that the author of "Indicator Practice and Steam Engine Economy" has prepared a catechism of the power plant, will be all the recommendation needed for this little book. It contains a great deal of condensed common sense about the management and operation of the steam plant. It is stated in the preface that it was at first intended to prepare a book that would contain just the information that an engineer should have in order to pass an examination for a license. From the fact, however, that no examiner is likely to ask any stated questions, the contents were made more comprehensive than was intended, and it is believed that the book will not only help those applying for a license, but will be of value to any seeking more general information upon steam engineering.

To enumerate the different subjects treated of would be to give a list of the apparatus and various points of management and theory that an engineer should know about to handle his plant intelligently. They include practical subjects, like pumping hot water, cleaning boilers, lining up an engine and pounding; subjects of economy, like feed-water heating, high pressure steam and condensation; and subjects upon general principles, like strength of boilers, action of steam in the cylinder, indicator diagrams and decimal fractions. A number of useful tables are appended.

SOFT COAL BURNING is the title of a pamplet written by C. M. Higginson, the fifth edition of which has recently been brought out by "The Railway Master Mechanic," 816 The Rookery, Chicago. The pamphlet discusses no patent schemes, but presents some of the general principles of combustion with a description of the simpler methods that may be used to relieve the smoke nuisance. It proceeds upon the assumption that if the trouble from smoky chimneys is to be abated, it must be by securing such perfect combustion in the furnace that the smoke will not be produced rather than by attempting to burn the carbon in the escaping gases after the smoke has once been formed. The pamphlet should be in the hands of all who are using soft coal.

ADVERTISING LITERATURE.

THE STANDARD SIZES FOR CATALOGS ARE 9 X 12, 6X9 AND 3½X6 INCHES. THE 6X9 IS RECOMMENDED, AS THIS SIZE IS MOST LIKELY TO BE PRESERVED.

A. D. QUINT, Hartford, Conn. Illustrated catalogue of Quint's Turret Drilling, Milling and Tapping Machines. 24 pages, 6 x 9 standard.

In these machines the principle of the turret lathe is applied in a way that results in the saving of much time on a class of work where slower methods in changing from one tool to another have always been in vogue. On the drilling machines pieces can be drilled and tapped without moving, and drills of different sizes, machine reamers, or other tools may be brought to bear upon the work in succession. A special reversible tap holder is used. The milling machine made upon this principle has four spindles working from one center, and four cutters can be used upon the same piece in succession.

THE WEBSTER & PERKS TOOL CO., Springfield, O., have sent us a circular of their new improved self-oiling and polishing lathe, and also of their 1-inch bench grinder. The buffing lathe has several new features.

THE NEW BRITAIN MACHINE CO., New Britain, Conn. Catalogue of the Case Automatic High-Speed Engine. 39 pages, illustrated. 6 x 9 standard.

The Case engine is well known in the electric lightng field.